Vol. 17, No. 10 October, 1961

LUBRICATION ENGINEERING

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

METAL BELLOWS SEALS

See Page 470

JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS

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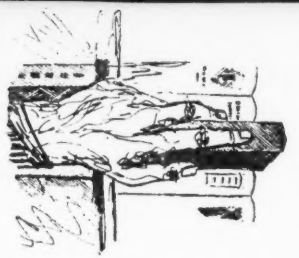

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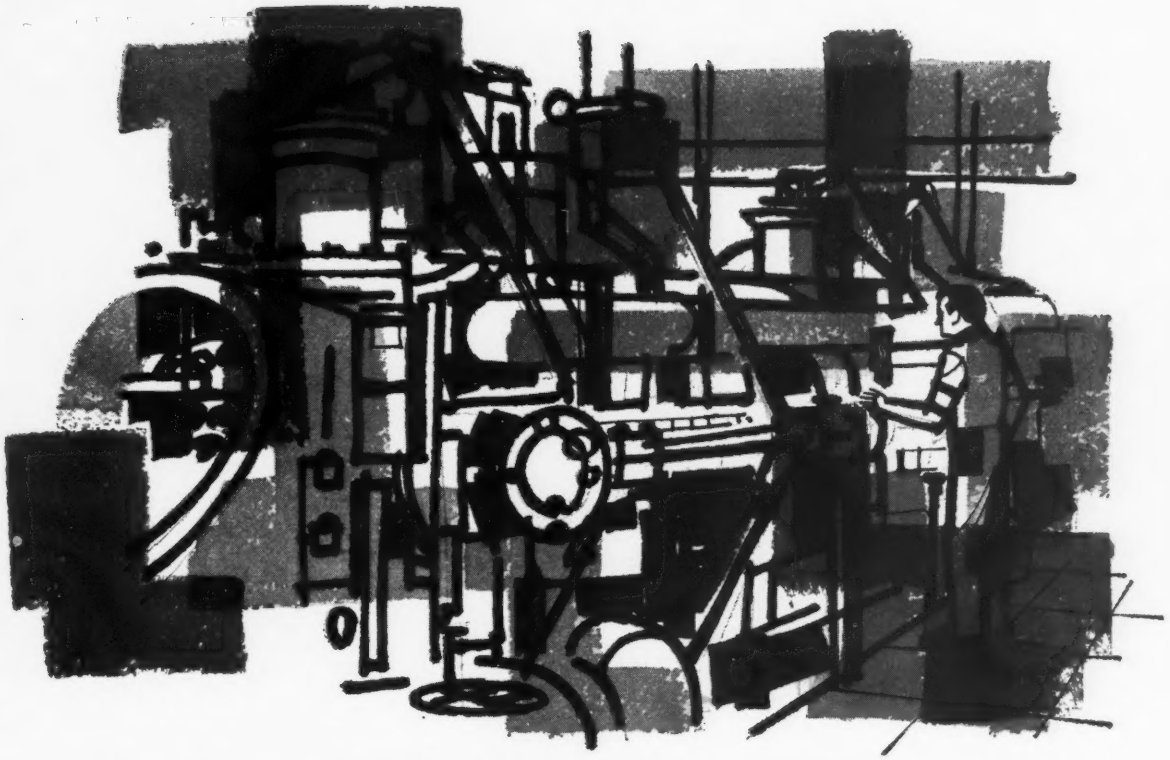

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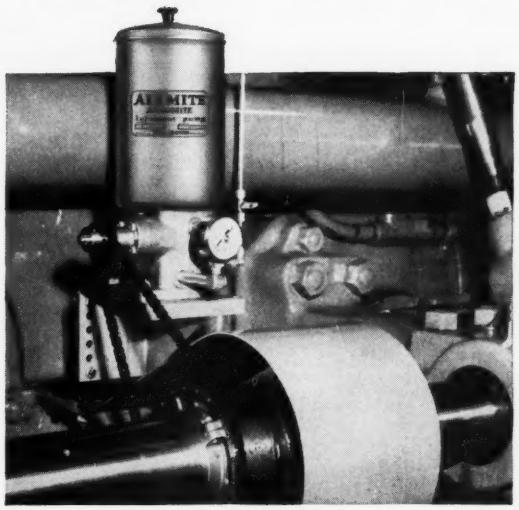

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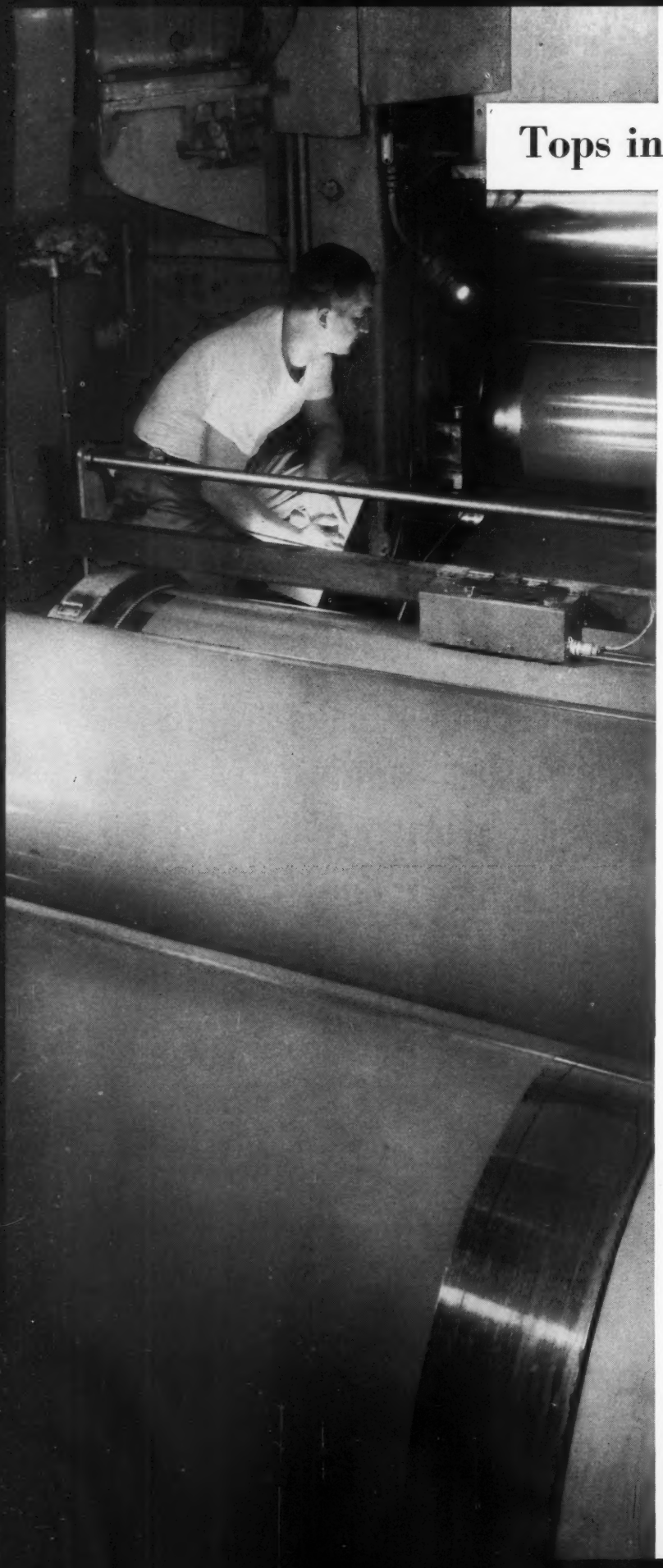

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Volume 17, No. 10
October, 1961

Lubrication Engineering

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

Contents

LUBRICATION IN PRACTICE
Metal Bellows Seals ....R. W. Blair, D. L. Johnson, and J. P. Morley 470
Use of Aluminum in Lubrication Systems—Design and Specifications .....E. E. McDole 488

RESEARCH, DESIGN, DEVELOPMENT
Materials for Elevated Temperature Piston Ring and Seal Ring Applications (600 F-1200 F) .....G. F. Hyde and J. H. Fuchsluger 476
A Study of the Torque Characteristics of Grease-Packed R-2 and R-3 Ball Bearings .....J. C. Lawrence 484

CURRENT NEWS
Lubrication in the News 453
Letters to the Editor 457
Personals 457
New Products 458
Product Literature 463
Society News 465
Section News 467
Order Form for ASLE 1961 Lubrication Conference Papers 468

EDITORIAL TOPICS
ASLE Is Yours 469

LITERATURE SUMMARIZED
Lubrication Abstracts 494
Patent Abstracts 495

TRADE INFORMATION
Advertisers Index 496

Abstracted by Engineering Index and Industrial Arts Index

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LUBRICATION IN THE NEWS

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PROCEEDINGS OF THE FIRST INTERNATIONAL SYMPOSIUM ON GAS-LUBRICATED BEARINGS NOW AVAILABLE

Sponsored by the Office of Naval Research and conducted by the Franklin Institute, this International Symposium on Gas-Lubricated Bearings was held in Washington, D. C. in 1959.

The Symposium, now available in printed form, provides realistic guidelines for continuing research, serves to attract the additional efforts of competent scientists in related areas of research, and assists in publicizing the vital contributions which these bearings have already made, and promise to extend, in highly diversified fields of application.

This 617-page volume records the 20 technical papers presented at the Symposium, in their entirety, as well as final panel discussion. A bibliography of some 290 entries appears as an appendix in this volume, which is currently available from the Superintendent of Documents, Government Printing Office, Washington, D. C. Clothbound, $4.25 per copy. (Source: Franklin Institute, Philadelphia, Pa.)

GULF OIL COMPLETES LUBE OIL UNIT

Gulf Oil Corporation has announced the completion of a new lubricating oil unit at its Port Arthur, Texas, Refinery. The 13,000 barrels-a-day installation will utilize a new lube oil hydrofinishing process developed by Gulf Research and Development Company, Gulf Oil's research subsidiary.

The new unit is Gulf's first venture into the field of hydrofinishing lube oils and a Gulf-developed solid catalyst is the heart of the once-through process. At present, the refinery plans to operate with a minimum of five days between feed stock changes.

The principal advantages of the new system are said to be: lower processing cost, higher yields, and better product. The new unit will produce a stable lube oil since nitrogen compounds, unsaturated hydrocarbons, sulphur, and naphthenic acids are removed to a higher degree than heretofore achieved. In some cases, yields presently are 4 to 20% higher than the refinery's past experience with clay contacting or acid treating.

To insure constant quality control, exact duplication of the daily operation is performed at the refinery's experimental laboratory on a bench-size model. In addition, special operating problems and conditions are simulated by the laboratory and examined well in advance of the actual needs of the unit. (Source: Gulf Oil Corp.)

SIMPLIFIED SUPPLY AND MAINTENANCE OF AIR-TRANSPORTABLE NUCLEAR POWER PLANT WITH LUBRICANTS FOR SUB-ZERO OPERATION

The first air-transportable nuclear power plant will soon go into operation exposed to high winds and subzero temperatures near Sundance, Wyoming, with minimum possibility of supply and lubrication difficulties. This is the type of formidable operating problem that plants will endure when producing power and heat in the depths of the Arctic and Antarctic.

Designed for service in the most inaccessible areas of the earth, PM-1 (portable medium power) was built by Martin for the Air Force under contract with the U.S. Atomic Energy Commission. The entire plant has been air lifted in sections by Air Force C-130 transports to Ellsworth Air Force Base, South Dakota. From there, the packaged nuclear plant was trucked to the site near Sundance, Wyoming.

Temperatures there go down to 40° below zero Fahrenheit and wind velocities reach peaks as high as 150 mph. In addition, there is an average of 110 in. of snowfall per year. PM-1 has been designed for operation under conditions that are even more severe than those at Sundance.

The portable nuclear reactor unit is of the pressurized water type, with a core containing about 64 lb. of uranium 235 dispersed in the walls of hundreds of steel tubes. The fuel core is good for two years before replacement. This unit will supply 1,000 kilowatts of electrical energy for operating radar equipment and 7 million Btu's per hour in steam to heat Air Force buildings.

In a cooperative program, the Martin Company's Nuclear Division, designer and builder of the portable nuclear plant, and Shell Oil Company have carefully chosen lubricants which will assure sustained operations under these extreme climatic conditions. Supply and lubrication problems have been simplified through selection of only three oils and one grease which will assure sustained operations under these extreme climatic conditions.

By its role as lubrication consultant on this project, Shell Oil Company has helped to simplify operational problems of PM-1 while assuring sustained operation with minimum attention to maintenance. (Source: Shell Oil Co.)

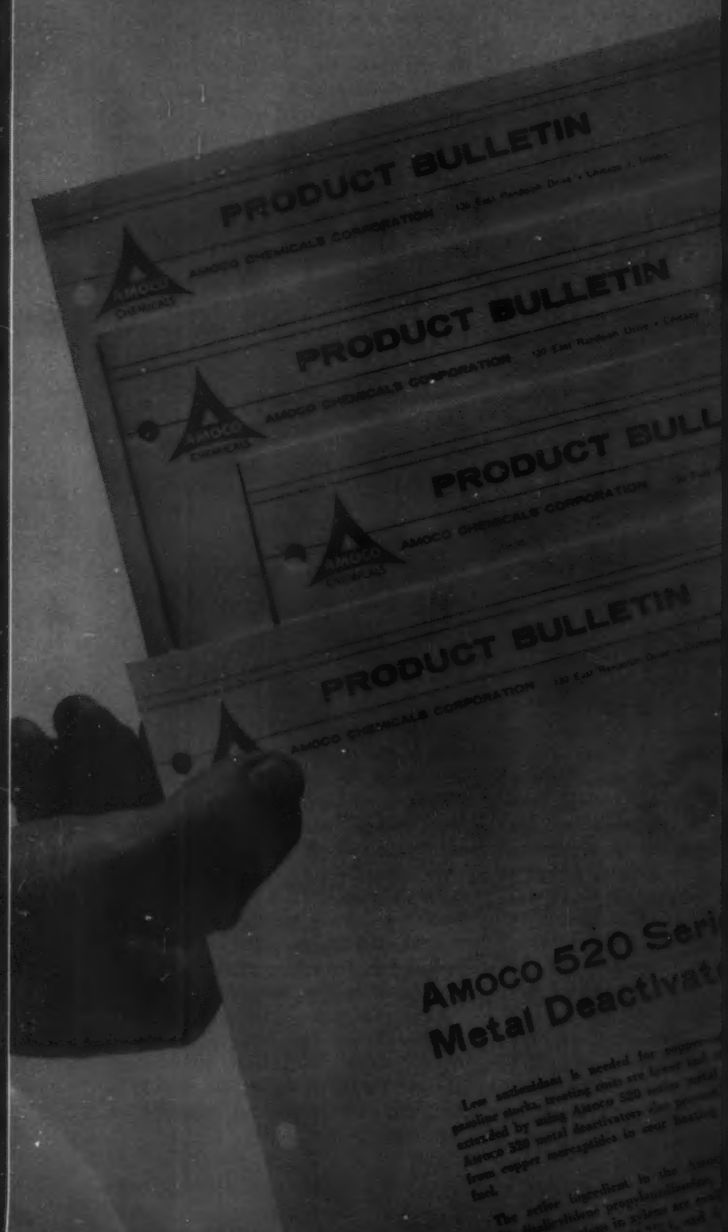
ASTM STANDARDS ON PETROLEUM PRODUCTS & LUBRICANTS D-2 — VOLUME II COMPILED

The American Society for Testing Materials announces that it has published a 2nd volume to the compilation of ASTM Standards on *Petroleum Products & Lubricants D-2*. While this volume includes a number of standards which are related to the petroleum field, it should be understood, however, that this compilation does not contain all the standards in the related fields. Those standards included in the compilation are directly related to the petroleum industry.

Copies of this publication may be secured from the American Society for Testing Materials, 1916 Race Street, Philadelphia 3, Pa. at $7.50 per copy. (Source: American Society for Testing Materials)

Perhaps you can solve a Petroleum Additive problem right here

AMOCO Chemicals offers an extensive line of additives for the upgrading of motor oils, gasolines and other fuels. At right are quick facts about the principal additives in the line. If you should like additional information, check the coupon below. We'll give your request immediate attention.



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LETTERS TO THE EDITOR

Lubrication Engineering welcomes comments and suggestions of readers with regard to articles published in its pages or concerning topics of interest to those engaged in any phase of lubrication engineering. Letters received cannot be acknowledged, but those felt to be of interest to other readers will be published. The Society will not be responsible for the statements or opinions expressed.

Sir:

We were quite interested in the father and son write-up covering the Forsythes in the July issue of LUBRICATION ENGINEERING especially due to the fact that both Senior and Junior are personal friends.

You mention "as far as can be discovered, this is one of the few father and son membership combinations in ASLE." We are proud to advise that there is another such combination—the writer and his son Warren who have been very active in the Buffalo Section; the writer, one of the early founders of the Buffalo Section, and Warren, now acting as Treasurer. Warren is Vice President and Technical Director of this Corporation and we merely pass this along for your information.

G. W. Miller, President
Battenfeld Grease & Oil Corp. of New York

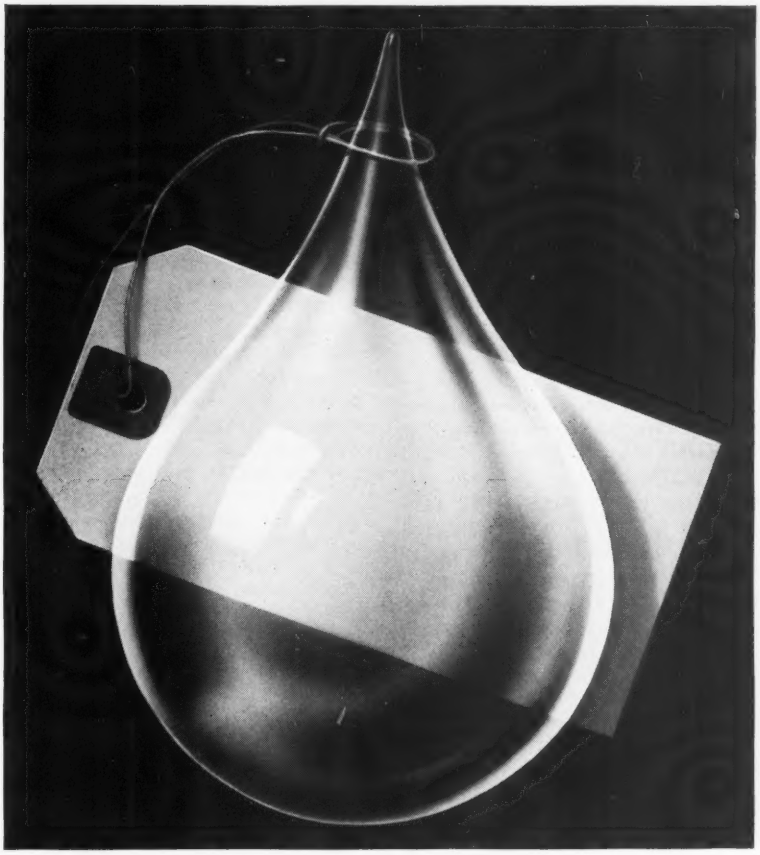
PERSONALS

The appointment of **A. C. Metz, Jr.** as local Field Staff Engineer for its Technical Service Division has been announced by Sinclair Refining Company. Mr. Metz will represent Sinclair's lubricating sales department with the automobile companies and equipment manufacturers in the Detroit area.

The appointment of **Frank J. Bednarczyk** to the post of chief quality engineer has been announced by SKF Industries, Inc.

James W. Hopkinson, ASLE President in 1955-56, died on September 18, following a long illness. Mr. Hopkinson was active in ASLE for many years and was recognized throughout the industry as one of the Deans of Lubrication Science. His many friends throughout the industry mourn his passing.

What's the PRICE TAG



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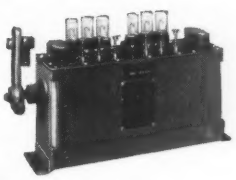

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NEW PRODUCTS

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NEW LUBRICANT ADDITIVE ANNOUNCED FOR ROLLING ALUMINUM

An oiliness additive, Nopco 9390, which is said to prevent breakdown of normal lubricants during high-speed, high-pressure rolling of aluminum sheet and foil has been announced by Nopco Chemical Company, Newark, N.J.

The additive—Nopco 9390—is useful in the drawing, extruding, stamping and machining of aluminum. The ingredients in the additive comply with requirements of the U.S. Food and Drug Administration, and Nopco 9390 is generally recognized as safe for use in food wrappings.

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SINGLE AND MULTIPLE SIGHT FEED VALVES AVAILABLE

Oil and other liquids can be fed by gravity or pressure from dispensers through these valves to lubricate bearings and for other purposes. This new design incorporates a sediment chamber which traps all dirt and grit to assure safe and dependable lubrication. They are especially recommended for installation where space is limited, for hazardous or hard-to-get-at locations. The valves are available from Trico Fuse Mfg. Co., Milwaukee, Wisc. in single or multiple outlets up to 24. Each outlet is provided with a sight feed port and has an independent vibration and leakproof friction needle adjustment which allows feed regulation from a steady flow to complete shut-off.

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SYNTHETIC COOLANT FOR GRINDING AND HONING

A fluorescent green aqueous compound is now available for grinding and honing ferrous alloys, but should not be used on aluminum or copper base products. 601 Synthetic Cooling Compound, when replacing conventional oil coolants, is said to generally alleviate process cleaning in washers following the machining operation. The manufacturer, Baker-Gubbins Co., Detroit, Michigan, states that the product contains rust inhibitors and will not deposit salts in use dilutions of 50:1.

Circle No. 3 on Reader Service Card

NITROHARD DEVELOPED TO PREVENT GALLING OF METAL SURFACES

A liquid salt bath nitriding process, called "Nitrohard," to prevent galling of metal surfaces, has been developed by W-K-M division of ACF Industries, Inc., in cooperation with the E. F. Houghton Company, Philadelphia, Pa.

With the new process, which makes use of an additive that provides an accelerator to increase the availability of nitrogen to steel, the manufacturer states that the following advantages may be noted:

1. Minimized dragout loss due to the lower viscosity of the molten salt.
2. Markedly improved heat distribution throughout the molten bath.
3. Easier cleaning of parts following completion of the process.

Circle No. 4 on Reader Service Card

NEW PACKING CONSTRUCTION FOR MOLYBDENUM DISULPHIDE

Molybdenum disulphide is now formulated for a mechanical packing construction. Moly-Pak is square braided asbestos yarn thoroughly impregnated with a special grade molybdenum disulphide. Moly-Pak is available in sizes ⅛" up to 1" on 2½ pound spools and in coil form 5 and 10 pounds approximate weight boxes. It is non-abrasive, non-corrosive and the lubricant has remarkable adherence to metal surfaces that provides long trouble-free service. Moly-Pak is recommended for centrifugal applications handling steam, water, oil, caustics.

Circle No. 5 on Reader Service Card

GREASE GUN WITH BUILT-IN LIGHT UNIT

A high pressure grease gun with built-in light has been developed by Product Engineering Company, Portland, Oregon. Named the LubriLite, the new gun makes it possible to save grease and time often wasted on shadowy, hard-to-see lube fittings.

A self-contained battery powered spotlight operates by the same short-stroke trigger that fires the lubricant. Because of its self-contained battery and non-sparking metal construction, the gun can be operated safely even when highly inflammable materials are present.

Circle No. 6 on Reader Service Card

ALL PURPOSE LOOB INTRODUCED

Spray Products Corp., Camden, N. J., announces the addition of Spray White All Purpose Loob to the firm's line of Spray products.

Spray White All Purpose Loob in the 16 oz. pressurized can was developed to provide clean lubricant where needed. Spray White All Purpose Loob will protect all metal surfaces and lubricate hard to reach areas.

Circle No. 7 on Reader Service Card

(*Continued on page* 459)

NEW PRODUCTS

(*Continued from page* 458)

KARBATE IMPERVIOUS GRAPHITE GLOBE VALVE AVAILABLE IN 1-INCH SIZE

A new 1-inch Karbate impervious graphite globe valve is available from National Carbon Company, Division of Union Carbide Corporation, New York City.

In this Type G valve, corrosive fluids contact only Karbate impervious graphite and "Teflon" plastic, eliminating the possibility of metallic contamination. Flanged connections make installation simple, and a cast iron casing protects the valve from mechanical damage and piping strains. The hand wheel rotates on lubricated ball bearings that lock it to the yoke, and an indicator shows valve position.

Recommended maximum operating conditions are 100 psig and 338 F. The valves can be operated in horizontal, vertical, or 45-degree positions. A sliding, non-rotating spindle facilitates sealing of the valve packing, and makes the valve adaptable to motorized, automatic operation.

Circle No. 8 on Reader Service Card

SELF-LUBRICATING CHAIN NOW AVAILABLE IN 1" PITCH

The Whitney "MSL" (maximum service life) line of self-lubricating power transmission and conveyor chains are now available in 1" pitch, American Standard No. 80, it was announced today by the Whitney Chain Co., subsidiary of Foote Bros. Gear & Machine Corp.

The expanded line of 1" Pitch No. 80 MSL Chain is available in single, double, triple and quadruple strands, in riveted or cottered type. Average ultimate strengths range from 13,000 lbs. in single strand to 52,000 lbs. in quadruple strands.

In operation, the lubricant contained within the bushing expands and flows over all vital bearing surfaces. When the drive stops, the lubricant contracts and is reabsorbed by the bushing. This cycle continues throughout the chain's service life. No additional oil is required.

Circle No. 9 on Reader Service Card

FILTERED GREASE PACKAGE

Bearing Inspection, Inc. of Huntington Park, California announces the availability of high/low temperature ball and roller bearing grease filtered to 10 or 45 microns. Grease is supplied in ½ ounce syringes, with needles, as a standard package unit, and will meet or exceed most "Mil Spec" requirements.

It is felt that the primary application of the product is in the regreasing of permanently sealed bearings. However, many greasing problems currently existing in miniature and other precision bearings also can be readily solved. The syringe packaging is designed to eliminate handling contamination. Since the filtering greatly reduces the abrasive qualities normally found in most "Mil Spec" greases, considerable expense can be eliminated due to the resulting extended bearing life.

The packaged grease is available in unfiltered form for other industrial applications where only handling is a problem.

Circle No. 10 on Reader Service Card

NEW EXTREME LOW TEMPERATURE SILICONE RUBBER MARKETED

A new extreme low-temperature resistant silicone rubber compound designed for use in molded and extruded seals and gaskets and a wide variety of mechanical applications, has been developed by the Silicone Products Department of General Electric, Waterford, New York.

Designed SE-5401U, the new material is a gray, 40 durometer compound which may be used to meet specifications MIL-R-5847D, Class I, Grade 40 and AMS-3334.

SE-5401U is an easy processing compound with typical properties after 24 hours/480 F.

Circle No. 11 on Reader Service Card

NEW WATER SOLUBLE CUTTING FLUID ANNOUNCED

Van Straaten Chemical Company, Chicago, has announced a new water soluble cutting fluid for use in metal cutting operations.

This new specially formulated water-soluble compound is said to have effected cost reductions in numerous metal cutting and grinding operations, and promises to be capable of replacing straight oils in about 95% of the operations still requiring straight oil compounds. The new formula is known by the number "Vantrol 700".

Because of its lubricating properties, plus its wetting and cleaning abilities, the compound is offered for use in broaching, form grinding, milling, drilling, hobbing, tapping, gear shaping and slotting operations where before only a straight oil would work.

The compound mixes readily in either soft or hard water. It offers good resistance to rancidity, will provide long coolant life and, under normal operating conditions, will afford good rust inhibition.

Circle No. 12 on Reader Service Card

NEW TROLLEY CONVEYOR LUBRICATOR

A new, portable, air-powered trolley conveyor lubricator that is easily moved from one conveyor to another is now offered by the Aro Equipment Corp., Bryan, Ohio. Weighing only 26 pounds, the new portable trolley conveyor lubricator can be installed by one man, even while the conveyor is running, and will lubricate from 100 feet to 100 miles of conveyor on one filling of lube.

The unit consists of a mounting frame that is welded to the existing conveyor rail; right and left hand lubricator sections, air and lubricant supply lines and a lubricant pump mounted on a mobile cart to provide floor-level control. Installation, according to the manufacturer, is fast and easy. The Aro portable trolley lubricator can be installed permanently in a conveyor line.

Circle No. 13 on Reader Service Card

NEW INDUSTRIAL LIQUID FILTER DEVELOPED

The filter unit consists of a splashproof motor, direct connected to a 35 GPM centrifugal pump, with an oversize debris strainer on the inlet. Tank contains an extended surface type throwaway filter element capable of removing particle sizes down to 5 microns, and special elements are available for removal of even smaller particles. Applications include industrial water, cutting fluids, soluble coolants and many other types of liquids.

This industrial liquid filter is offered by The Hilliard Corp., Elmira, N. Y.

Circle No. 14 on Reader Service Card

Purchasing Agent, ". . . purchasing fewer types of lubricants in greater volume lowers our costs."

Superintendent, ". . . simplifying anything, including lubrication, pays off in improved efficiency."

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In some plants, Gulf engineers have found more than *three* times the number of lubes actually needed for efficient operation. This means three times as many ordering, billing, stocking and handling problems. And it also means three times as many chances for human errors.

Examples: One large electrical appliance manufacturer slashed lubricants from dozens to just 6. In addition, a well-known paper company pared lube brands from 39 to 13.

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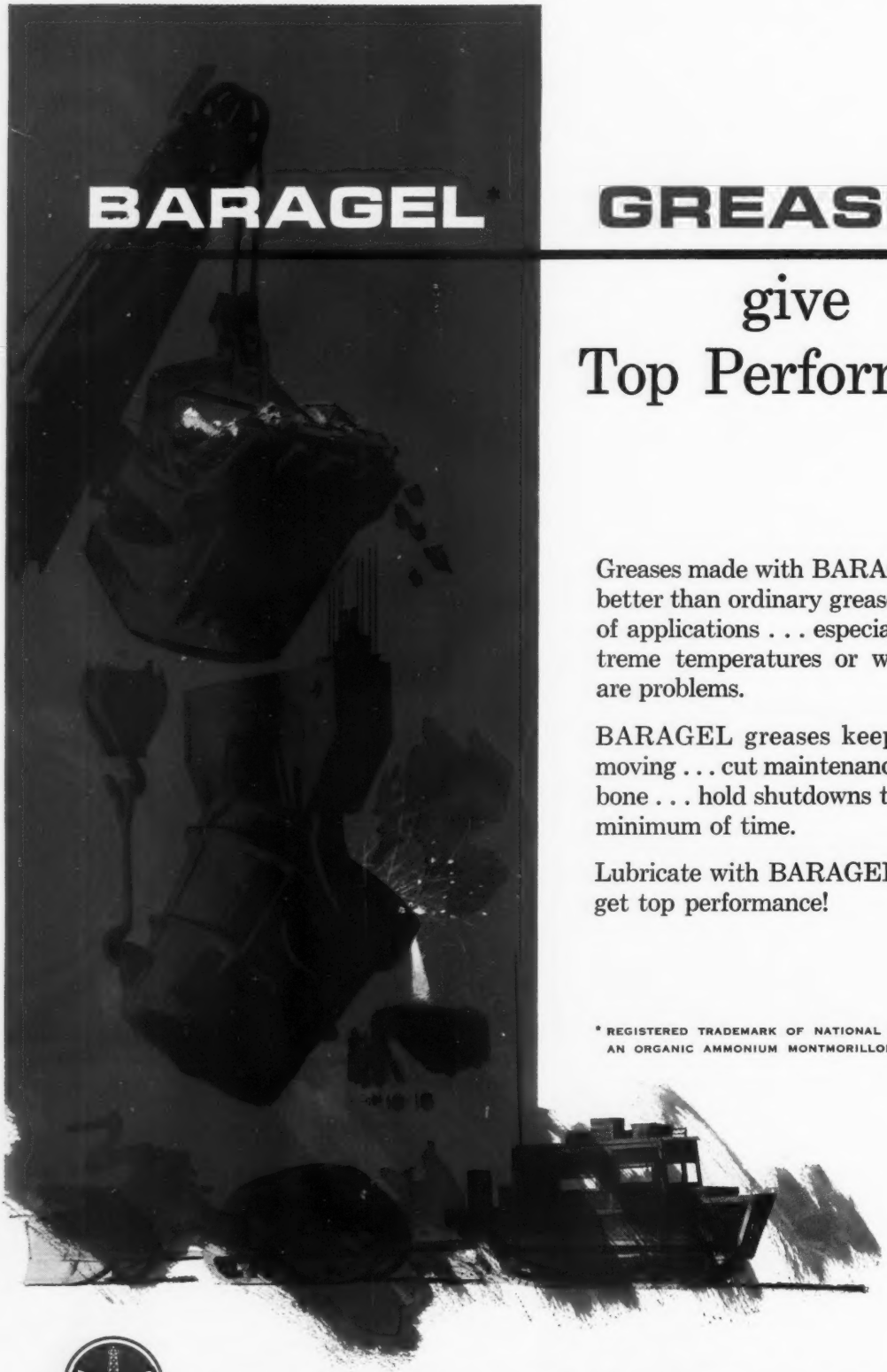

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PRODUCT LITERATURE

The Society will not be responsible for the accuracy of statements made in this column. Any reference to proprietary products does not imply an endorsement of such products by the Society.

CATALOG ON PNEUMATIC PRODUCTS AVAILABLE

An entirely new catalog covering the complete line of pneumatic products from C. A. Norgren, Englewood, Colo., is now available. Catalog No. 1000 contains 64 pages describing compressed-air-line filters, pressure regulators, air-line lubricators, valves and other miscellaneous products.

In addition, there is a section devoted to useful engineering information such as a compressed-air flow chart, moisture content of saturated, compressed air, dimensional data, etc.

Circle No. 15 on Reader Service Card

ACCUMULATORS GUIDE PUBLISHED

Hydraulic Accumulators, their operating principles, designs and uses are described in a new edition of "Guide to Hydro-Pneumatic Accumulators" by Greer Hydraulics, Inc., Los Angeles, Calif.

The manual is prepared for design engineers and others interested in hydraulic circuitry and equipment. It contains details of the operations of all types of accumulators including piston, mechanical bellows, spring loaded, aircraft and bladder types.

Circle No. 16 on Reader Service Card

NEW TECHNICAL REFERENCE ON SILICONE FLUIDS

The Silicone Products Department of General Electric offers a new, comprehensive technical reference which describes the broad range of major silicone fluids used as hydraulic and damping fluids, dielectrics and lubricants, antifoam agents, coatings for ceramics and glass and as additives in urethane foams.

Designated S-9, the 20-page, two-color publication differentiates between available grades of silicone fluids, according to viscosity and classifies them on the basis of physical properties. An index permits the user to select a particular fluid on the basis of properties required in his application. The publication is illustrated with photos, curves and tables, providing application ideas as well as pertinent data.

Circle No. 17 on Reader Service Card

NEW BULLETIN DESCRIBES FILTER ELEMENT

Accurate, simple and quick measurement of the largest hole in a filter element can be made with the Bubble Point Test Stand described in Bulletin A11, available from Aircraft Porous Media, Inc., a subsidiary of Pall Corporation, Glen Cove, New York. The test stand is used to detect damaged or otherwise imperfect elements prior to installation in an aircraft, missile or ground system.

The bulletin includes specifications and application data. It describes two types of stands available, one using hydraulic fluid and the other using denatured alcohol as the test medium. Both types contain all necessary controls, as well as an air filter to guarantee that contaminants will be excluded from the item under test. All instructions for operation are included on the control panel of the unit. Detection range of the instrument is from 13 to 120 microns.

The bulletin also includes a photograph of the test stand with several standard adaptors, and a schematic drawing of the instrument. Military and industrial technical manuals on the Bubble Point Test Stand are also available on request.

Circle No. 18 on Reader Service Card

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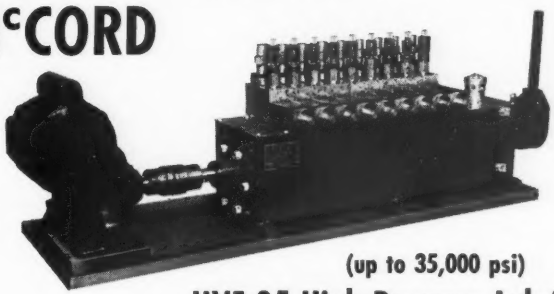

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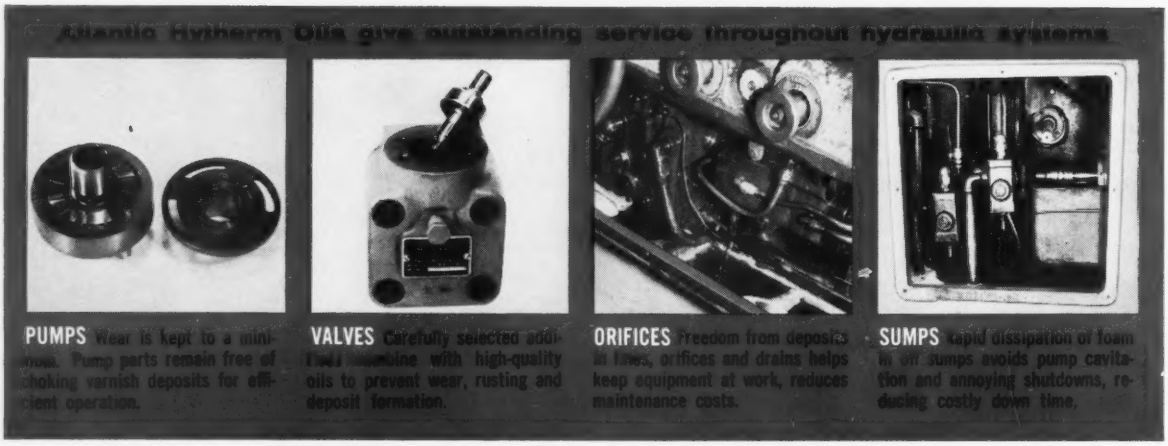
Atlantic Hytherm Oils give outstanding service throughout hydraulic systems
PUMPS Wear is kept to a minimum. Pump parts remain free of choking varnish deposits for efficient operation.
VALVES Carefully selected additives combine with high-quality oils to prevent wear, rusting and deposit formation.
ORIFICES Freedom from deposits in lines, orifices and drains helps keep equipment at work, reduces maintenance costs.
SUMPS Rapid dissipation of foam in oil sumps avoids pump cavitation and annoying shutdowns, reducing costly down time.

The type of fluid power transmission medium used in a hydraulic system is largely responsible for its economical and effective operation. Realizing the importance of oil stability to successful service, manufacturers of hydraulic equipment long have recommended oils having the same properties found in turbine lubricants which must operate continuously for long periods of time.

Atlantic Hytherm Oils are basically turbine quality products. They are produced from solvent treated paraffin base stocks which have a high viscosity index. In addition, they are fortified with additives to resist oxidation and foaming as well as to protect machine parts against rust.

The most outstanding feature of Hytherm oils is their anti-wear characteristic . . . a direct result of this unique additive combination. Compared in performance under identical conditions, with high-quality straight mineral oils of comparable viscosity, Atlantic Hytherm Oils actually cut wear-rate of vane type pump parts 100 times. Savings in maintenance costs and equipment down time far exceed the moderate increase in cost of the Hytherm oils above ordinary mineral oils.

And Atlantic Hytherm Oils also provide excellent versatility. In fact, they've been used successfully in a variety of "hot-oil" heat exchange systems with a minimum of undesirable deposits. They are highly recommended for lubricating compressor cylinders, rotary compressors, enclosed gear and chain drives, dryer roll circulating systems, machine tool drives, small steam turbines and other applications where over-all stability is highly necessary.

Your Atlantic Sales Engineer will welcome the opportunity to discuss the application of these oils to your specific requirements. In cases where detailed analysis is required, the facilities of the Product Development & Technical Services Division are at your disposal, entirely without obligation. Simply call or write to The Atlantic Refining Company, Pennypacker 5-2345, 260 S. Broad Street, Philadelphia 1, Pa.

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SOCIETY NEWS

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Crystal Lake, Illinois

Leon E. Ericson
Material Service Div.
General Dynamics
Chicago, Illinois

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Bristol, Connecticut

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Acheson Colloids Company
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YOUNGSTOWN

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Youngstown Sheet & Tube
Youngstown, Ohio

Robert T. Davidson (R)
Sun Oil Company
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Warco Equipment Company
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San Antonio, Texas

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Shell Oil Company
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Caterpillar Tractor Company
Peoria, Illinois

Melvin G. Burdett
Caterpillar Tractor Company
Peoria, Illinois

Robert D. Clevenger (R)
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Chicago, Illinois

Harry E. Davis
Caterpillar Tractor Company
Peoria, Illinois

Gale H. Ekstrand
Caterpillar Tractor Company
Peoria, Illinois

William T. Fall
Caterpillar Tractor Company
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James L. Hopper
Caterpillar Tractor Company
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Robert G. Krueger
Texaco, Incorporated
Peoria, Illinois

Ralph M. Kurz
Texaco, Incorporated
Peoria, Illinois

Walter A. Milkereit
Caterpillar Tractor Company
Peoria, Illinois

B. Miskus
Caterpillar Tractor Company
Peoria, Illinois

George W. Murray
Mechanical Rubber & Supply Co.
Peoria, Illinois

Paul P. Prunch
Mechanical Rubber & Supply Co.
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Walter Norris Engineering Co.
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George R. Runyan (R)
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E. Peoria, Illinois

John P. Shubert
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Frank Taylor
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J. William Vollentine, Jr.
Caterpillar Tractor Company
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A. A. Wald
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Sealed Power Corporation
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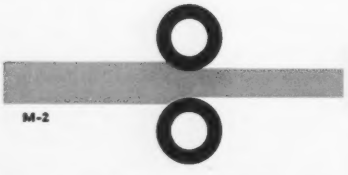

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SECTION NEWS

Edwin Martin Strom, Jr., has been elected chairman of the New York City Section. Mr. Strom is manager of the graphite and lubricants division of the Joseph Dixon Crucible Company, Jersey City, N.J.

Rochester—Following a meeting of the Section Officers and Board of Directors on September 13, members and guests of the Rochester Section held their September meeting. Mr. Harold W. Whiting, Manager Quality Service, Compressor and Engine Div., Worthington Corp., Wellsville, N. Y., surveyed present lubrication practices of both rotary and reciprocative compressors. Present trends in synthetic oils were also reported.

Boston—Dr. William Campbell will discuss "Solid Lubricants" at the November 20 Section meeting.

Los Angeles—"Molybdenum Disulfide as a Grease Additive," was discussed at the September meeting by James T. McCabe and William M. Stilwell, Climax Molybdenum Company.

Baltimore—It has been mutually agreed that Section Meetings would be held on the 2nd Thursday each month in lieu of the 3rd Thursday to avoid conflict with the Philadelphia Section Meetings. "Lubrication Features of New Bar Mill," will be discussed at the November meeting by Bob Ulrich, Steelton Plant, Bethlehem Steel Company.

Philadelphia—The Philadelphia Sections of ASLE and ASME (Power Fluid Division) will combine in a joint meeting on October 19, 1961. The program will include two speakers. Mr. Robert G. White, Vickers, Inc., Philadelphia, will discuss "Variable Speed Hydraulic Drives". Mr. Richard J. Koerwer, Mobil Oil Company, Philadelphia, Pa., will discuss "Hydraulic Fluids". The next ASLE Philadelphia Section meeting is scheduled for November 16, 1961.

Last year's Los Angeles Section Chairman, L. P. (Rick) Richardson (right), who was formerly with Richfield Oil Corp., and is now associated with The Lubrizol Corp., Cleveland, Ohio, is shown passing the gavel to this year's Section Chairman, P. A. (Pete) Sorenson, Richfield Oil Corp., Pacific Palisades, Calif.

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☐ "Bearing and Lubricant Requirements for Some Aero-Space Projects: Discussion and Preliminary Investigation," by R. H. Larson and A. G. Piken, Bendix Research Laboratories, Southfield, Mich. *(ASLE Paper No. 61LC-1)*

☐ "Friction, Wear, and Evaporation Rates of Various Materials in Vacuum to 10^{-7} mm Hg," by D. H. Buckley, M. Swikert and R. L. Johnson, NASA, Lewis Research Center, Cleveland, Ohio. *(ASLE Paper No. 61LC-2)*

☐ "Friction and Wear Characteristics of Cermets at High Temperature and High Vacuum," by R. D. Brown, R. A. Burton and P. M. Ku, Southwest Research Institute, San Antonio, Texas. *(ASLE Paper No. 61LC-3)*

☐ "Friction and Surface Deformation During Sliding on a Single Crystal of Copper," by J. M. Bailey, Beloit College, Beloit, Mich., and A. T. Gwathmey, University of Virginia, Charlottesville, Va. *(ASLE Paper No. 61LC-4)*

☐ "Chemical Changes in Steel Surfaces During Extreme Pressure Lubrication," by D. Godfrey, California Research Corp., Richmond, Calif. *(ASLE Paper No. 61LC-5)*

☐ "Extreme Pressure Lubrication and Wear—The Chemical Reactivity and the Extreme Pressure Action of Two Aliphatic Disulfides," by A. Dorinson and V. E. Broman, Sinclair Research Laboratories, Harvey, Ill. *(ASLE Paper No. 61LC-6)*

☐ "Polymer Formation on Sliding Metals in Air Saturated with Organic Vapors," by W. E. Campbell, Consultant, Cleveland 6, Ohio, and R. E. Lee, Jr., General Electric Co., Schenectady, N.Y. *(ASLE Paper No. 61LC-7)*

☐ "Effect of Lubricant Viscosity on Metallic Contact and Friction in a Sliding System," by M. J. Furey and J. K. Appeldoorn, Esso Research and Engineering Co., Linden, N.J. *(ASLE Paper No. 61LC-8)*

☐ "Heat of Preferential Adsorption of Surfactants on Porous Solids and its Relation to Wear of Sliding Steel Surfaces," by A. J. Groszek, British Petroleum Co., Ltd., Sunbury-On-Thames, Middlesex, England. *(ASLE Paper No. 61LC-9)*

☐ "Preparation, Properties, and Some Applications of Super-Refined Mineral Oils," by E. E. Klaus, E. J. Tewksbury and M. R. Fenske, Pennsylvania State University, University Park, Pa. *(ASLE Paper No. 61LC-10)*

☐ "Dynamic and Static Irradiation of Nuclear Power Plant Lubricants," by D. B. Cox, E. A. Oberright and R. J. Green, Socony-Mobil Oil Co., Paulsboro, N.J. *(ASLE Paper No. 61LC-11)*

☐ "Viscous Lubrication in Wire Drawing," by J. F. Osterle and J. R. Dixon, Carnegie Institute of Technology, Pittsburgh, Pa. *(ASLE Paper No. 61LC-12)*

☐ "Effects of Base Oil Viscosity and Type on Bearing Ball Fatigue," by F. G. Rounds, General Motors Corp., Warren, Mich. *(ASLE Paper No. 61LC-13)*

☐ "The Effect of Contact Angle on Rolling Contact Fatigue and Bearing Load Capacity," by E. V. Zaretsky, W. J. Anderson and R. J. Parker, NASA, Lewis Research Center, Cleveland, Ohio. *(ASLE Paper No. 61LC-14)*

☐ "Weibull Distribution of Rolling Contact Fatigue Life and Deviations Therefrom," by T. Tallian, SKF Industries, Inc., Philadelphia, Pa. *(ASLE Paper No. 61LC-15)*

☐ "Evaluation of Ball Bearing Performance in Liquid Hydrogen at DN Values to 1.6 Million," by H. W. Scibbe and W. J. Anderson, NASA, Lewis Research Center, Cleveland, Ohio *(ASLE Paper No. 61LC-16)*

☐ "Lubrication of Rolling Surfaces by a Ree-Eyring Fluid," by J. C. Bell, Battelle Memorial Institute, Columbus, Ohio. *(ASLE Paper No. 61LC-17)*

☐ "The Design of Externally Pressurized Gas Thrust Bearings for Maximum Stiffness and Stability," by D. C. Allais, IBM Research Laboratory, San Jose, Calif. *(ASLE Paper No. 61LC-18)*

☐ "Pressure and Flow Studies of an Experimental Externally Pressurized Gas Lubricated Bearing," by G. B. Speen, ITT Federal Laboratories, San Fernando, Calif. *(ASLE Paper No. 61LC-19)*

☐ "Analysis and Design of Externally Pressurized Gas Bearings," by I. Tang and W. A. Gross, IBM Research Laboratory, San Jose, Calif. *(ASLE Paper No. 61LC-20)*

American Society of Lubrication Engineers
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Donald B. Sanberg,
Executive Secretary, ASLE

METAL BELLOWS SEALS

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A metal bellows can be incorporated in an end face type seal. Temperature extremes or gamma radiation dictate the use of a metal bellows in such a seal. When able to calculate the spring rate of the bellows, the seal designer can set the proper sealing face load. The bellows seal design is not complete until the effects of temperature, pressure, and mating ring wobble are taken into full account.

Nomenclature

C=Constant
D=Mean Diameter of Bellows, In.
E=Modulus of Elasticity, psi
A=Effective Area, in.^{2}
H=Bellows Wall Thickness, in.
L=Radial Width of Convolution, in.
N=Number of Convolutions
p=Pressure, psi
R=Spring Rate per Convolution, lb/in/conv.
Sp=Stress Due to Pressure, psi
$SCFM$=Standard Cubic Feet per Minute
W=Load, lb

INTRODUCTION

A liquid fuel missile contains two of the most difficult sealing applications in existence today—the cryogenic liquid pump seal and the hot turbine gas seal. Coupled with the temperature extremes in these two applications will be found high surface speeds and pressures. End face seals are usually selected for these difficult jobs.

An end face seal can be designed to incorporate a metal bellows and is known as a bellows seal. This type of seal is designed for applications where it is desirable to eliminate the usual elastomeric or plastic internal seal. Fig. 1 shows a bellows seal that has been sectioned. The carbon sealing ring is shrunk into the top carrier. A welded metal bellows is welded to the top carrier and the bottom housing.

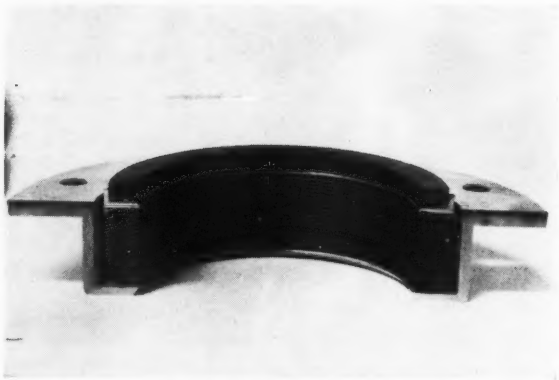

Fig. 1. Cross section of normal welded bellows.

The principal operating condition that usually dictates the use of a bellows seal is temperature—either high (500F and above) or low (—65F to absolute zero). Another operating condition that dictates the use of a metal bellows is the presence of gamma radiation. Gamma rays will normally have deleterious effects on most polymers.

Added advantages of a metal bellows are the elimination of elastomeric or plastic internal seal friction and the elimination of internal locking devices in the seal. As a result, the sealing face is able to follow mating ring wobble and axial shift with much less frictional resistance.

Presented at the 16th ASLE Annual Meeting in Philadelphia, Pa., April 1961

The purpose of this paper is to acquaint the seal designer and seal consumer with the various parameters associated with metal bellows as used in end face seals. It is also intended to point out some of the advantages and disadvantages of the two basic types of metal bellows. With this information, the designer should be in a better position to choose between a metal bellows seal or some other type of end face seal for a given application.

BELLOWS FABRICATION

Basically, there are two types of metal bellows, the hydraulically formed type and the welded type.

The hydraulically formed type is made by drawing a tube of suitable metal into a continuous number of convolutions to form the complete bellows. The resultant bellows convolutions are "U" shaped and require a large amount of axial space. While the method of manufacture is comparatively economical in large quantities, a high tooling cost is involved.

The welded bellows type consists of formed plates or diaphragms which are joined together in a controlled atmosphere welding operation. This type of bellows offers the greatest choice of convolution shapes to meet a large variety of mechanical properties, e.g., low rate, high burst pressure, high endurance limit, and others. The axial spacing or pitch can be extremely small, often permitting a bellows seal envelope to be as small or smaller than that of a standard face seal. Extensive tooling is also required for this bellows.

BELLOWS SEAL DESIGN CONSIDERATION

In order to properly design a metal bellows seal, the designer must know how to handle the basic bellows parameters. Such parameters as the metal's modulus of elasticity, diaphragm size and shape, and the number of convolutions have a direct effect on the spring rate of the bellows.

Spring Rate:

Spring rate may be defined as the force required to compress or extend the bellows one unit of length (usually expressed as pounds per inch). If a seal designer wants a face load of 10 pounds with a bellows seal that has a spring rate of 50 pounds per inch, he has to design for .200 inch compression at the operating height. A 200 pound per inch rate in the above case would require a deflection of only .050 inch to attain the 10 pound face loading.

As shown above, the designer can work with a wide range of spring rates and arrive at a given face loading by changing the compression of the bellows. The compression can be varied by changing the free height of the bellows for a given operating height.

The lowest spring rate compatible with all other design considerations is desired for it must be kept in mind that the face seal must work throughout an axial operating range of .020 inch to .125 inch or more. An operating range of .125 inch with a bellows seal having a spring rate of 100 pounds per inch—a typical value —would have a 12.5 pound load variation. This variation could be far in excess of the load bearing capacity of the sealing face materials on a small seal. Such a small bellows seal, possibly having .40 square inch sealing face area and designed for a 20 psi mean face pressure with an operating range of ±.062 inch, would have a load variation of 1.8 to 14.2 pounds or over 700% from minimum to maximum operating height.

Several technical papers have been written on various bellows convolution configurations by Feely and Goryl[1], Hetenyi and Timms[2], and Turner and Ford [3]. A general formula for spring rate takes the form:

$$R = \frac{CD^xEH^y}{NL^z} \qquad [1]$$

The exponent "x" normally varies between 1 and 2, and the exponents "y" and "z" vary between 2 and 3 for various designs. It can be seen from the above, that the seal designer can juggle several parameters in order to obtain a desired bellows spring rate. Since welded bellows permit a wide choice of convolution shapes, the designer can make the most of the formula. The typical variation in spring rate with respect to mean diameter for a given type of metal, welded convolution shape and three metal thicknesses is shown in Fig. 2.

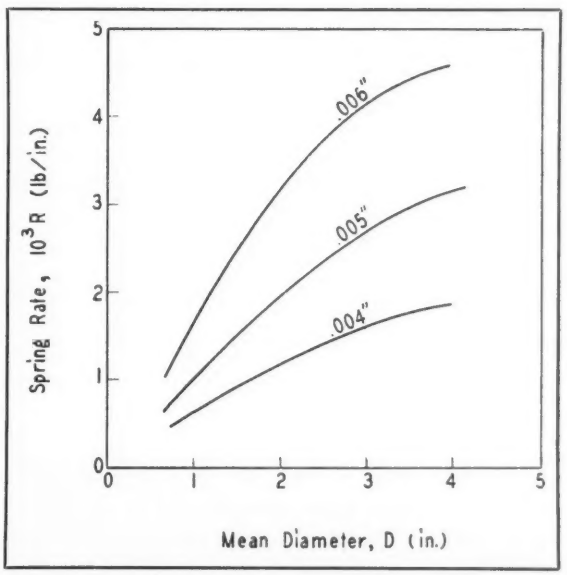

Fig. 2. Metal bellows spring rate variations with respect to mean diameter for various metal thicknesses.

It must be kept in mind that for some of the common stainless steels used in bellows (i.e., 300 series), the modulus of elasticity "E" can also be variable due to changes in temperature. Measuring a bellows load in a tensile tester while immersed in liquid nitrogen (—320F), showed an 11% increase over that measured at room temperature (Fig. 3). This increase in load was due to an increase in the

modulus of elasticity. At elevated temperature, the modulus can decrease by as much or more. With this particular material, the spring rate had decreased by 8% at 350F.

A tensile tester can be used to measure the load-deflection characteristics (spring rate) of bellows. The load-deflection chart of a bellows seal is shown in Fig. 4. It can be seen that there is a small amount of hysteresis and a linear load-deflection relationship throughout the entire operating range. The load-deflection chart, Fig. 5, of a standard end face seal with an "O" ring internal seal of similar size shows a good deal of hysteresis loss and an irregular load-deflection relationship.

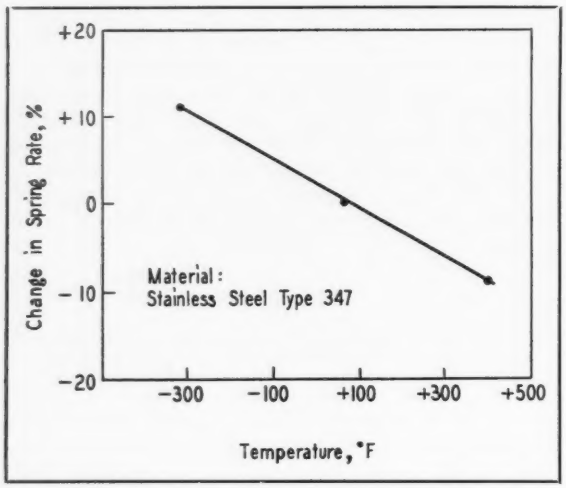

Fig. 3. Change in spring rate of welded metal bellows with temperature.

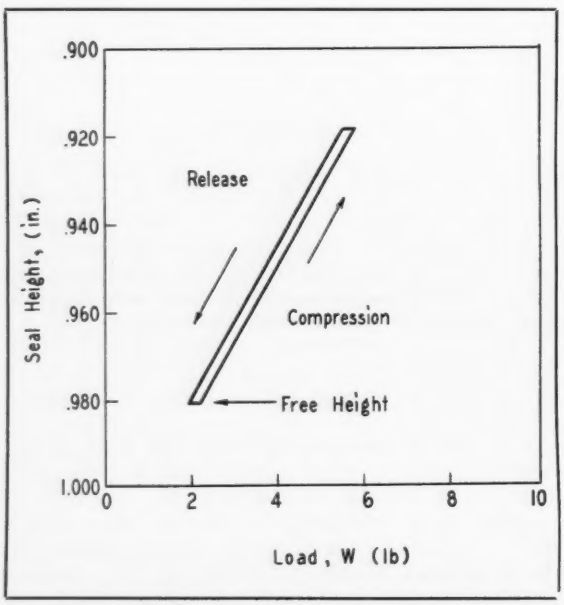

Fig. 4. Load-Deflection curve for welded metal bellows seal.

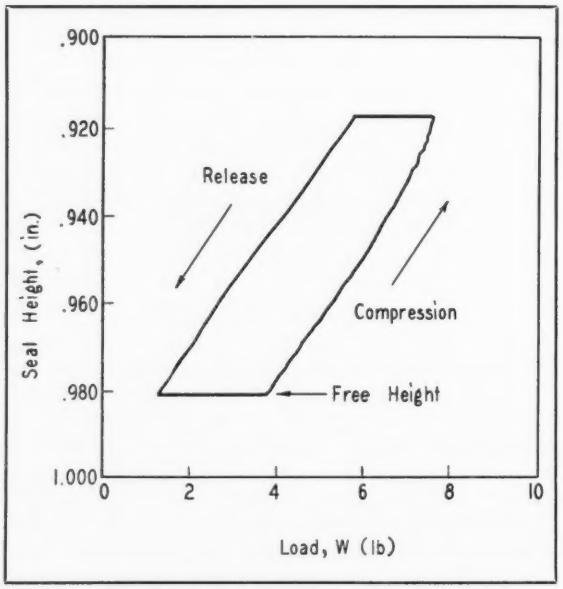

Fig. 5. Load-deflection curve for standard end face seal.

It is not enough for the designer to select a bellows design on the basis of best spring rate alone. His design must be able to withstand the maximum operating fluid pressure at the maximum operating temperature with a reasonable margin of safety, must have a satisfactory fatigue life if pressure fluctuations are present or mating ring wobble is significant, and should be free from resonance during operation.

Pressure Limits

Feely and Goryl have considered the bellows diaphragm to be a flat plate with fixed ends and derived the formula:

$$Sp = \frac{PL^2}{2H^2} \qquad [2]$$

While the above formula has to be modified for various configurations, it does give insight into the operating fluid pressure limit of a bellows. As can be seen, one can normally get a higher operating fluid pressure from a welded bellows for a given spring rate and envelope size (more convolutions per inch of axial space permits the use of a greater metal thickness for a given rate).

Fatigue Properties

It can also be seen from Eq. [2] that if the operating pressure is variable, the stress due to pressure loading will vary in direct proportion. The fatigue life then becomes an important consideration. Mating ring wobble will also produce a small change in stress in the bellows. The fatigue life of welded metal bellows was determined in order to predict the life expectancy of these bellows under actual operating condi-

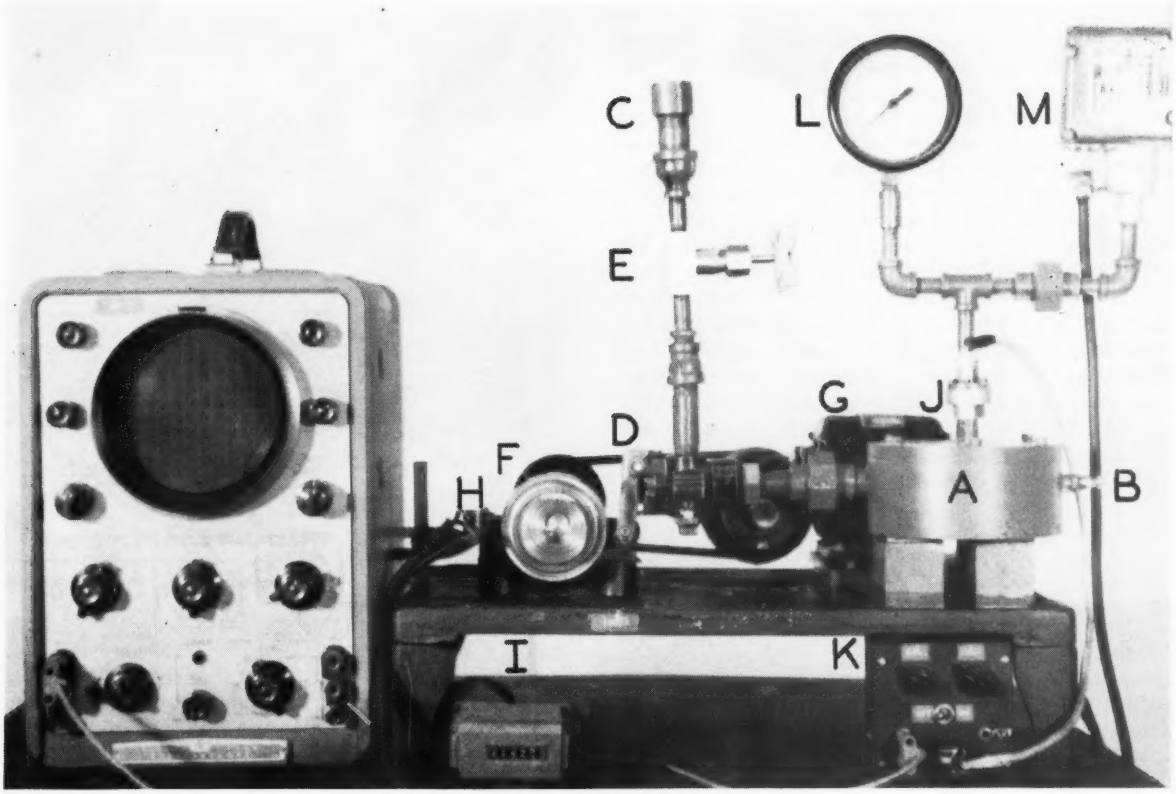

Fig. 6. Fatigue tester.

tions. The fatigue tester used for determining the fatigue life was basically a device for subjecting the OD (outside diameter) of a bellows to air pressure which varied between 90 psig and 60 psig, 230 times per minute. The ID (inside diameter) was kept at atmospheric pressure at all times.

The fatigue tester (Fig. 6) consisted of a chamber (A) in which the metal bellows undergoing test was kept compressed at its operating height. Air entered at 90 psig through inlet (B) and was exhausted through (C) via the cam operated valve (D) and the regulating valve (E). The air pressure acted only on the OD of the bellows. The cam (F) was driven by the motor (G). The cam also actuated the switch (H) of the counter (I) and thus an accurate record of the number of pressure cycles to which the bellows was subjected, was kept. The magnitude of the pressure changes inside the chamber was measured by a pressure transducer (J), the output of which was fed through the junction box (K) to the oscilloscope. The trace obtained on the oscilloscope is shown in Fig. 7. The pressure on the ID of the bellows was atmospheric and was measured by the gage (L). If failure of the bellows took place, the pressure would rise and show on the gage and at the same time, operate the pressure switch (M) which would stop the test. The machine could thus be operated unattended.

The results obtained in two specific instances are given in Table 1.

After these tests, each bellows was cut into eight sections and each section examined for cracks under a

TABLE 1.—Pressure Flex Test Results

TEST NO.	BELLOWS MATERIAL	THICKNESS OF DIAPHRAGM (IN.)	COMPRESSION FROM FREE HEIGHT (IN.)	NO. OF PRESSURE CYCLES	RESULT
1.	SS 347	.006	.030	4.1×10^6	No Failure
			.060	4.4×10^6	No Failure
2.	SS 347	.004	.040	15×10^6	No Failure

Bellows for Test No. 1: 1.562 inches OD x 1.062 inches ID, 9 Convolutions

Bellows for Test No. 2: 1.562 inches OD x 1.062 inches ID, 8 Convolutions

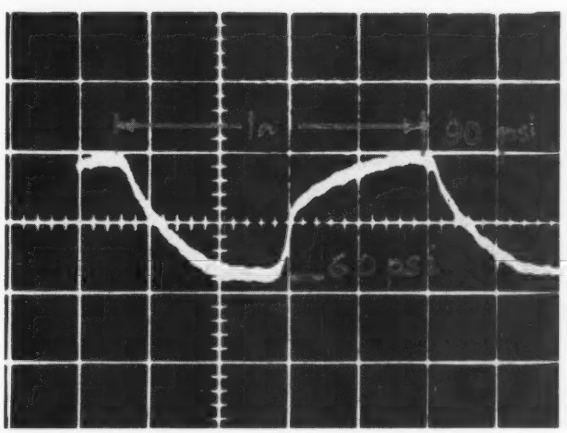

Fig. 7. Oscilloscope trace showing pressure changes inside the test chamber.

metallurgical microscope. They were also etched to make sure that the cracks were not being filled by flowed metal during sectioning and polishing. They were then ground down 1/16 inch, polished and re-examined. This was repeated for a third time. The total number of examinations was 24 (3 different places on 8 sections of the bellows). In no instance could any cracks be found. Fig. 8 shows one of the areas examined. The results of the metallurgical tests showed clearly that fatigue failure of this bellows had not even begun after 15 million cycles of operation.

Welded bellows and hydraulically formed bellows have also been subjected to severe flex tests. The bellows were clamped and flexed ¼ inch at the rate of 1720 cycles per minute by an eccentric. Formed bellows had circumferential cracks by 5 million cycles while welded bellows showed no signs of failure at 10 million cycles. Testing was discontinued at this point since it was only intended to give qualitative results.

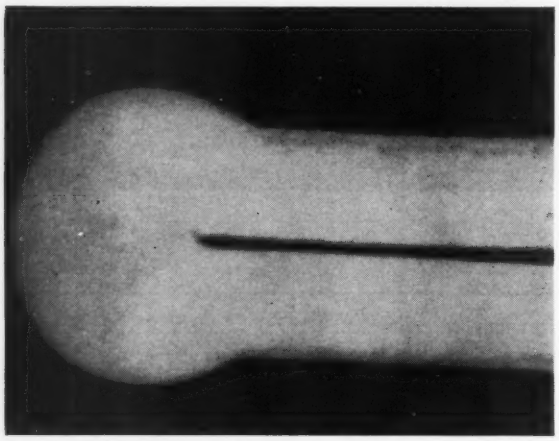

Fig. 8. Micro-photograph of two weld areas showing absence of any fatigue cracks (150x).

Seal Face Pressure Loading

If there is a pressure differential across the bellows, the piston effect must be taken into considerations. The area of this piston is known as the effective area and is approximately equal to the area corresponding to the mean diameter of the bellows.

$$A = \frac{\pi}{16} (D_{OD} + D1^{D})^2 \qquad [3]$$

The sealing face must be properly located or balanced[4] with relation to the effective diameter to prevent overloading or unloading.

The effective area for hydraulically formed bellows is fairly uniform for its working stroke; it can vary with pressure in welded bellows depending on the diaphragm design. Where constant effective area is required in welded bellows, as in control bellows or high pressure seals, the designer must select a diaphragm design that will give him a nearly constant effective area.

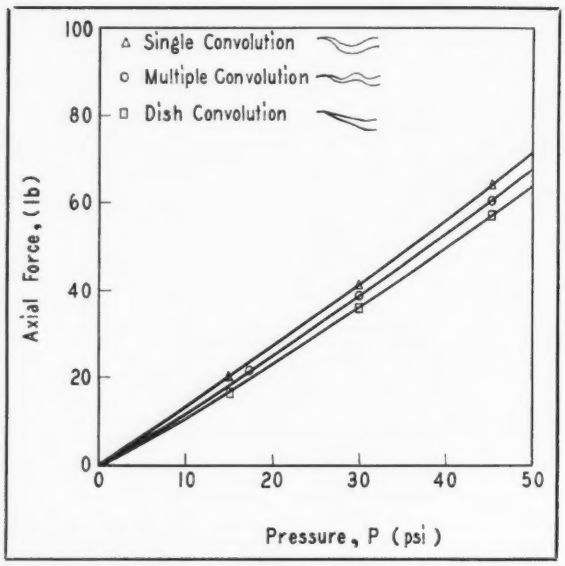

Fig. 9. Change in bellows axial force with internal pressure for three convolution shapes.

Fig. 9 shows the variation of axial force with pressure for several welded bellows configurations of the same size. The slope of the curves at any point is the effective area for that pressure. The single convolution design is nearly a straight line function (Force = A x p) while the multi-convolution and the deep-dish slopes have smaller and variable effective areas.

Seal Vibration

In a bellows seal, the seal designer must not overlook the effects of resonance. A bellows seal, operated at a resonant speed gave a 400% increase in air leakage at 5 psig; shortly thereafter, the carbon sealing face shattered. The designer can vary the mass of the bellows seal to move the critical speed beyond the maximum operating speed encountered in most applications. If this is not possible, every effort must be made to quickly pass through the critical speed or speeds during start-up or shut-down of the machinery containing the seal.

Bellows Metallurgy

When using metal bellows, the designer must consider certain aspects of the bellows metallurgy. The medium that is to be sealed, the pressure, the temperature, the spring rate, the fatigue life, and the fabrication must all be considered when selecting the materials of construction.

Bellows can be manufactured which will handle media varying from cryogenic liquids (*e.g. LOX, LH_2*), to liquid metals (*e.g. NaK, H_g*) and from hot gases (e.g. exhaust, air) to common oils (e.g. mineral oil, Mil-L-7808). The only limitation is that the corrosion resistant material be either weldable or formable.

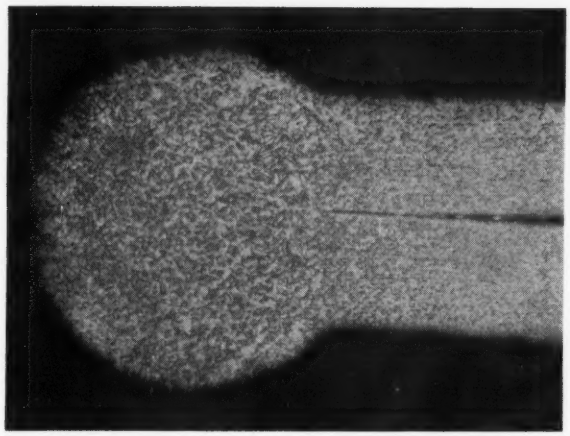

Fig. 10. Good weld properly heat treated. 10% $(NH_4)_2S_2O_8$ etch (200 x).

With the range of metals presently available, temperatures of —423F to 2000F have been handled. The strength of most metals at 2000F is relatively low so severe stress limitations must be imposed on this type of application. Pressures as high as 6000 psi can be sealed at 1600F providing the bellows diameter is small enough and a low spring rate is not critical.

The designer has at his disposal materials which vary in modulus of elasticity between 10×10^6 to 31×10^6 psi and yield strengths of 6000 psi to 200,000 psi. This great range of available metals is one of the many advantages of the welded bellows over the formed bellows. Due to forming difficulties, the material selection is more limited for formed bellows.

The importance of a sound weld that is properly heat treated cannot be over emphasized. Fig. 10 shows such a weld. Soltys [5], in his paper on handling liquid metals, points out that a properly heat treated welded metal bellows is the only satisfactory bellows for this application. It is felt that the following materials will handle most applications: Stainless steels 316, 321, 347, 17-7PH, AM-350, phosphor bronze, beryllium copper, aluminum 6061 T-6, titanium, corrosion and oxidation resistant nickel base alloys and age-hardenable nickel base alloys. Many of the other super alloys, high corrosion resistant materials and special alloys have been used. Before specifying any material, one should consider its physical and mechanical properties, weldability, formability, cost, corrosion resistance and heat treatability.

Welded joints are considered to have three zones which merge into one another. Those zones are the base or parent metal, the weld metal and the heat affected zone between the weld metal and base metal.

Great emphasis should be placed on the quality of the weld metal and the condition of the heat affected zone, since usually, the designer has selected his parent metal with great care. If the weld metal is of poor quality or if harmful notches occur between the weld metal and the parent metal or if the heat affected zone is not properly heat treated, difficulty may result from low toughness, low strength or low endurance limit. A thorough investigation must be made concerning the properties of the weld metal and the metal next to the weld. Samples should be tested to determine the proper heat treating procedure to be used on the finished parts. If these precautions are observed and if the material has been carefully selected, a welded metal bellows can be made that will perform well in practically all applications.

Sealing Face Materials

Severe conditions often require better sealing face materials than those commonly used in face seals. Testing of high temperature carbons, high speed tool steels, ceramics, carbides and cermets at elevated temperatures is necessary to find face materials that will keep pace with the bellows material capabilities.

One series of metal to metal sealing face tests run at 750F and 35,000 rpm (19,000 fpm) pointed out certain materials that gave low wear rates and air leakages. The leakage rate measured at 50 psig was less than 0.1 scfm which is comparable to that obtained from a standard face seal tested at 400F.

CONCLUSION

Metal bellows seals are being used to meet a wide range of operating conditions and consequently cover a broad price range. Developments in the past few years in high strength-high temperature alloys, welding techniques, and heat treating methods have greatly increased the reliability and range of operating conditions that can be handled by welded metal bellows seals.

By eliminating the internal friction and possible leakage path associated with the O-ring secondary seal or any other type of internal seal used in most conventional end face seals, the metal bellows seal has taken two steps toward more versatile, dependable seals.

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Materials for Elevated Temperature

PISTON RING and SEAL RING

Applications (600 F-1200 F)

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An introductory explanation of the basic functions of piston rings and seals is presented, together with a summary of the properties required in materials used for such parts. The increasing severity of operating conditions in various fields is pointed out. In this respect, increasing temperatures and reduced lubrication are cited as the principal factors that necessitate the development of improved piston ring and seal materials.

The approach to this problem was to screen a large number of proprietary materials by means of laboratory bench tests, followed by component testing of the more promising materials under simulated operating conditions. The materials considered can be divided into the following categories; Carbons; Carbides, Nitrides, Borides, and Intermetallics; Ceramics and Cermets; Irons and Steels; High Temperature Alloys; Hard Facing Alloys; Surface Treatments and Coatings. In general, the best materials from the standpoint of wear and oxidation resistance were found to be impregnated carbons, metallic carbides, and intermetallics. Unfortunately, most of these materials present problems of design and fabrication.

Presented at the 16th ASLE Annual Meeting in Philadelphia, Pa., April, 1961

INTRODUCTION

Piston rings, piston ring type seals, and shaft seals are employed for one basic purpose. This purpose is to provide a gas or liquid seal in a system of moving parts. In recent years conditions of heat and corrosion have become predominant considerations of the overall piston ring and sealing ring picture. This has necessitated extensive research in the field of materials to find or develop new materials which possess special unique properties to functionally withstand these severe conditions.

The work conducted at the authors' company over the past five years, done mainly under two WADC programs (1), has been directed primarily into the field of materials for high temperature wear applications, and the adaptation of these materials into functional piston ring and sealing ring parts. The properties of principal interest can be divided into the following general categories: Corrosion Resistance—the ability to withstand oxidizing or other corrosive gasses without appreciable change; Wear Resistance—low wear rates and wear surface compatibility at room and elevated temperatures (2) in the absence of conventional lubricants; Resistance to Stress Relaxation—the ability to retain the spring like properties required of a sealing member, even under conditions of extreme heat; Conformability—the ability of the sealing member to conform to the contour of its mating surface under conditions of varying temperatures; and Relative Ease of Fabrication—many of the newer materials, while promising as sealing members from a functional point of view, are presently outside the realm of consideration due to the cost involved in making them.

GENERAL APPROACH

The approach, to finding the materials that would have the necessary properties to meet these more severe operating conditions, was to survey the field of existing materials, to select from those that could possibly qualify, and then subject these to closer evaluation. This entailed several steps. Initially, a literature survey was performed to select categories of materials and typical materials within each category.

With the initial literature survey completed, the leading suppliers of the various categories of materials were contacted, more often than not, by visits to their plants. Tentative material selections were discussed with these people and, in the light of the combined knowledge of the intended application and the basic material properties, the original list of material selections was modified.

Basic material selections having been made, samples for screening evaluation tests were then obtained and subjected to laboratory bench type wear tests and laboratory oxidation tests. The materials showing promise in these tests were then fabricated into finished components and reevaluated under actual operating conditions. Methods of evaluation will be covered in detail later. With this beginning, contact was maintained with newer developments through a continuing literature survey and through continued contact with the material suppliers.

MATERIALS CONSIDERED

The materials considered fell into the following categories: Carbons (compounded and bonded electrographites, sometimes impregnated); Carbides, Nitrides, Borides, and Intermetallics; Ceramics and Cermets; Irons and Steels; High Temperature Alloys; Hard Facing Alloys; Surface Treatments and Coatings. A complete list of the materials investigated is presented in Table 1. The categories of Surface Treatments and Coatings became an important phase of work in themselves in that they represent a method of adding good wear properties to materials that are desirable in all other respects but are inherently poor wearing. Also, many good wearing materials that are available as surface coatings are prohibitive in cost and extremely difficult to fabricate as solid bodies. The field of coatings and surface treatments, applicable to piston rings and seals, will be developed more fully later in this paper.

METHODS OF EVALUATION FOR SOLID MATERIALS

The solid materials were evaluated for wear in two bench type wear tests. First, they were run on the wear test machine, shown in Fig. 1, at room temperature. Those which wore well in this test were then

TABLE 1.—MATERIALS INCLUDED IN THIS INVESTIGATION

CARBONS:
- Impregnated Carbon "A"
- Impregnated Carbon "B"
- Impregnated Carbon "C"
- Impregnated Carbon "D"
- Impregnated Carbon "E"
- Carbon "F"
- Carbon "G"

CARBIDES:
- Tungsten Carbide
- Titanium Carbide
- Chromium Carbide
- Silicon Carbide
- SiN bonded SiC
- SiC+Graphite
- Boron Carbide

NITRIDES:
- Boron Nitride
- SiN impregnated BN
- Phosphate bonded BN
- BN+Boric Acid
- BN+Graphite
- Silicon Nitride

BORIDES:
- TiB-CrB
- Cr-Mo-CrB
- Ni-Ag-WB-"A"
- Ni-Ag-WB-"B"
- Ni-Ag-MoB
- Ni-Ag-CrB

INTERMETALLICS:
- Titanium Aluminide
- Titanium Disilicide
- Nickel Aluminide
- Molybdenum Disilicide

CERAMICS:
- Aluminum oxide
- Glass ceramic
- Beryllium Oxide
- Synthetic Mica
- Silicon
- Fused Silica
- Hydrous Aluminum Silicate
- Phosphate bonded Talc

CERMETS:
- Cr-Al_2O_3
- Cr-Mo-Al_2O_3

STEELS:
- W-Mn-Mo Tool Steel
- M-3, Type 1 Tool Steel
- T-15 Tool Steel
- H-12 Tool Steel
- H-11 Tool Steel
- D-5 Tool Steel
- C-V Die Steel
- 316 Stainless
- 410 Stainless
- 17-4 PH Stainless
- 440 F Stainless
- Mod. 440 Stainless

HARD FACING ALLOYS:
- Ni-Cr-W-Fe
- Co-Cr-Ni-C
- Co-Cr-W-C

IRONS:
- Austenitic Gray Iron "A"
- Austenitic Gray Iron "B"
- Cr-C-Fe—"A"
- Cr-C-Fe—"B"
- Cr-Mo Gray Iron
- Austenitic Ductile Iron

HIGH TEMPERATURE ALLOYS:
- Cr-Mo-Ni Base
- Cr-Fe-Ni Base "A"
- Cr-Ni-Co-Fe Base
- Cr-Ni-W-Co Base "A"
- Cr-Ni-W-Co Base "B"
- Molybdenum
- Cr-Si-Ni Base
- Si-Cu-Ni Base
- Cr-Fe-Ni Base "B"
- C-Ni Base
- Cr-Co-Ni Base "A"
- Cr-Ni-Fe Base
- Cr-Mo-Co-Ni Base
- Cr-Ni-Co Base
- W-Cr-Co Base "A"
- Al-Mo-Fe Base
- Ni-Ag Base
- CdO-Ag Base
- Cr-W-Fe-Co Base
- W-Cr-Co Base "B"
- Cr-W-Co-Fe Base
- Ag impregnated Ni
- Cr-Co-Ni Base "B"
- Cr-Co-Mo-Ti-Ni Base
- Cr-Ni-Base

SURFACE TREATMENTS:
- Nitriding
- Malcomizing
- Sulfurizing
- Phosphating
- Oxidizing

METAL PLATINGS:
- Diffused Ni-Cd
- Sn Electroplate
- Ag Electroplate
- Cr Electroplate
- Sprayed Molybdenum
- Co-Ni Electroplate "A"
- Co-Ni Electroplate "B"
- Ni Chemical Plate
- Fe Electroplate

CARBIDE, OXIDE COATINGS:
- Co-WC—"A"
- Ni-WC-WCrC
- Co-WC—"B"
- Al_2O_3
- Ni-Cr-Cr_3C_2
- WC+Binder

MoS_2 COATINGS:
- MoS_2+Graphite+Resin
- MoS_2+W
- MoS_2+Resin
- MoS_2+Graphited Pb
- MoS_2

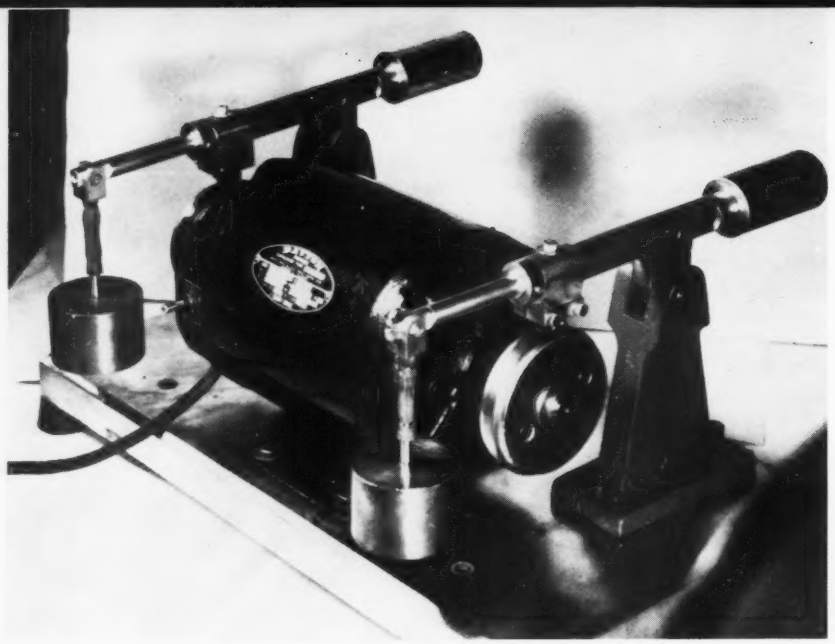

Fig. 1. Room Temperature Wear Test.

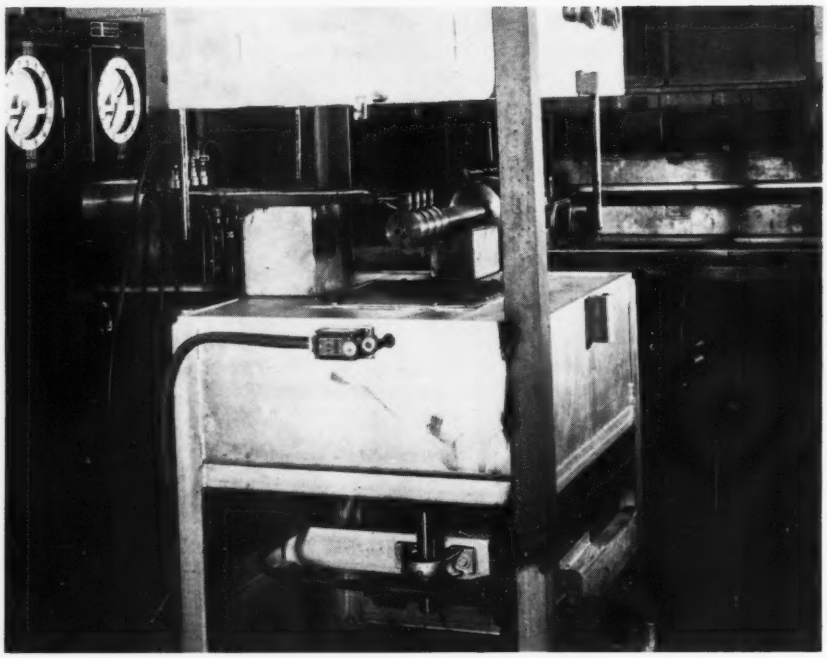

Fig. 2. Elevated Temperature Wear Test.

tested in an elevated-temperature wear test machine, shown in Fig. 2, at an ambient temperature of 1000 F. The principle of operation of both test machines is basically the same as shown in Fig. 3. The flat side of a block type rider specimen under a radial load of five pounds, and starting with line contact, is pressed against the periphery of a 3½ in. diameter ring. The ring rotates on a shaft driven by an electric motor to give a rubbing speed of 1,600 sfpm. The time of test is one hour, and no lubricant is used.

Specimens tested by this method are evaluated on the basis of overall wear rate and compatibility of the rubbing surfaces, ratings of "Good", "Fair", or "Poor" being given. A "Good" rating indicates relatively low wear rates and smooth, usually glazed, mating surfaces. "Fair" indicates a somewhat increased wear rate and some surface distress, such as slight scuffing or scoring. A "Poor" rating usually denotes high wear rate and catastrophic distress of the mating surfaces. Some typical wear tracks are illustrated in Fig. 4.

Since, as mentioned earlier, oxidation is one of the principal problems at elevated temperatures, all materials were also subjected to an oxidation test as part of their initial screening. The oxidation test is conducted in a circulating air furnace using a specially constructed fixture consisting of a number of triangu-

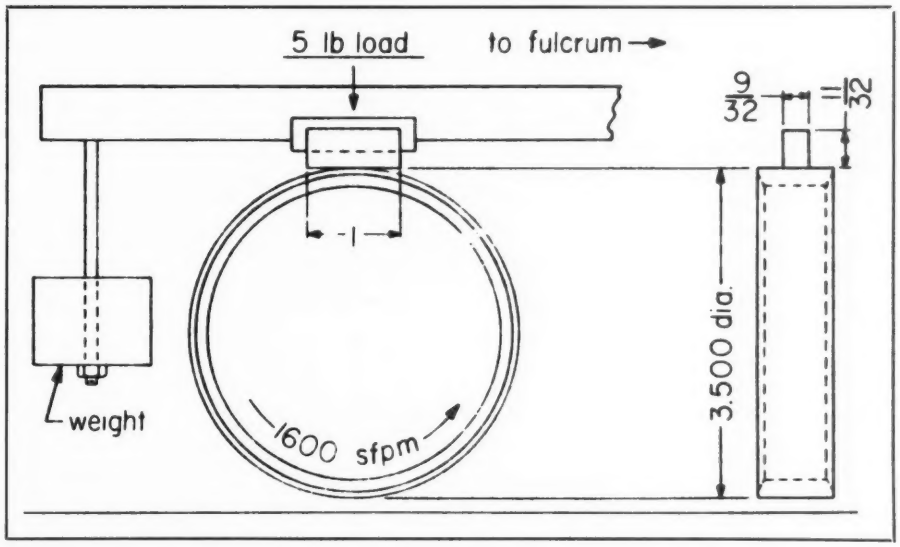

Fig. 3. Schematic Diagram of Wear Test.

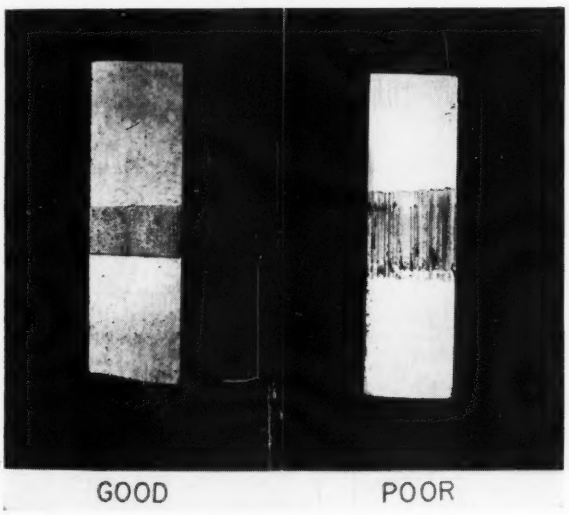

Fig. 4. Typical Worn Specimens.

lar screen trays arranged in tiers on a central spindle. Several specimens of each of many materials can be tested simultaneously.

The oxidation test consists of a series of heating and cooling cycles in an air atmosphere. The fixture containing the specimens is placed in a furnace which had been preheated to 1000 F, allowed to soak at this temperature for a noted length of time, and then removed to cool in still air. At intervals of 10, 100, 500, and 1000 hours of accumulated soak time for each material, specimens were removed for measurement, weighing, visual examination, hardness measurement, and microstructure examination. To be considered as having "Good" oxidation resistance, a material subjected to this test should remain virtually unchanged.

Materials rated as "Good" with respect to wear properties and oxidation resistance are then fabricated into piston rings or seals and tested in prototype engines. For this purpose the laboratory has a number of test units.

CONCLUSIONS REACHED WITH REGARD TO SOLID MATERIALS

Due to the large number of materials evaluated, the conclusions reached will be presented in summary form, and only typical wear results obtained with the more promising materials cited.

Carbons:

Most unimpregnated carbons distintegrated completely or deteriorated rapidly at temperatures of 1000 F. The impregnants developed in recent years, however, gave extended life at 1000 F to a number of carbons, and a few even show appreciable life at 1200 F and 1300 F. This is illustrated by the oxidation curves, shown in Fig. 5, for three representative im-

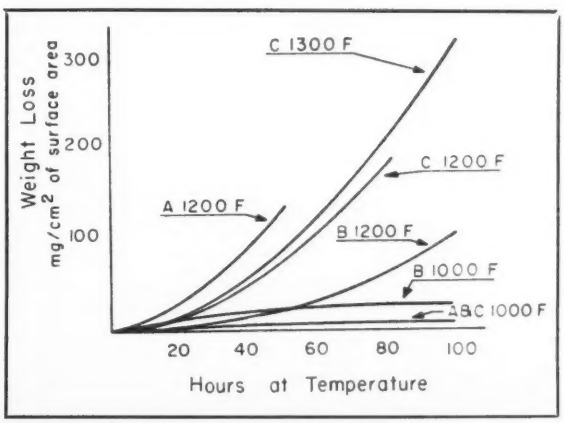

Fig. 5. Oxidation Rate of Carbon Materials.

TABLE 2.—Solid Materials—Room Temperature and Elevated Temperature Wear Ratings vs. a Variety of Mating Surfaces

MATERIAL	MAXIMUM SERVICE TEMP. OXIDIZING ATMOSPHERE (°F)	WEAR RATING*														
		CR C	TI C	WC	CR-Al_2O_3	CR-MO-Al_2O_3	Al_2O_3	CR PLATE	CR-NI-W-CO-"B"	CR-C-FE-"B"	NIT. NIT. 135 M	AISI 410	AISI 440 F	17-4 PH	AISI T-15	AISI M-3 TYPE 1
IMPREGNATED CARBONS:																
Carbon A	1100				F	G		G	G	G/G		F/F				
Carbon B	1200				G	G		G	F	G		F		F		
Carbon C	1300		G				G		G	G				P	G	
CARBIDES, BORIDES, NITRIDES, AND INTER-METALLICS:																
TiC	1100	G/G	G/G				G/G	P	P	F/G	F		P			
CrC	1300	G	G/G	G			G/P	P	F		F	P		P	G/G	F
WC	900		G/G	G/G			G/G	P	P		G	G	P	F	G/G	G	G/P
$MoSi_2$	1300	G/G	P			F	G		P	F						
SiNi	1300	G/G	G				G		F	P						
TiB—CrB	1200	G	F		P		G/G		P	G						
CERAMICS:																
Al_2O_3	>1300	G/G	G	G/G		P	G/G		P	P	F					
HIGH TEMPERATURE ALLOYS:																
Cr-Ni-W-Co-"A"	>1000	G/F			P		G/G		P							
Cr-Ni-W-Co-"B"	>1000	G/G	G/G	G/G	P	P	G/G	P	P	P	F	P			G/G	F
IRONS AND STEELS:																
AISI T-15	1000	G	G/G	G/G			G	G	F	P	F	P	P		G/G	G
C-V Die Steel	900		G				G	G	F						G	
AISI H-12	1000		G	G			F	F	F				P	P	F	
AISI 440 F	1000		G	G/G			G/F	P	F	P			P	P	G/G	
440 Modified	1000		G/G				G	G	G/P	P				P	G	

*G = Good, F = Fair, P = Poor; x/y x = Room Temp. Rating; y = 1000 F Rating.

pregnated carbons. Referring to Table 2 it can be seen that the impregnants do not appear to impair the excellent wear properties normally associated with these materials.

Carbides, Nitrides, Borides, and Intermetallics:

All of these materials resisted oxidation well at 1000 F. Some of these materials, of course, will withstand temperatures well in excess of 1000 F. As a class they are relatively good wearing, as can be seen from the data in Table 2. The metal carbides in particular show exceptionally good wear properties.

Ceramics and Cermets:

All the ceramics and cermets tested successfully withstood the oxidation test, as would be expected. These materials, too, are capable of withstanding temperatures considerably above 1000 F. The cermets as well as most ceramics tested show high friction and tend to score their mating surface. Aluminum oxide is one of the better wearing ceramics as is illustrated by the data shown in Table 2.

High Temperature Alloys:

High temperature alloys for the most part have excellent strength and oxidation resistance at 1000 F, but relatively poor wear properties. Only two materials, Cr-W-Ni-Co alloys, were found that showed "Fair" to "Good" wear properties, and these were against carbide and oxide mating surfaces.

Irons and Steels:

The best wearing, of the irons and steels evaluated, are those containing carbides. The data in Table 2 shows that the 440 type stainless steels and the tool steels, all of which have high carbide contents, have relatively good wear resistance against a variety of mating surfaces. The oxidation resistance of these materials, however, is not the best. The more oxidation resistant steels, such as the 300 series stainless and related alloys, have very poor wear properties.

Summation:

In general, the best materials from the standpoint of wear and oxidation resistance are high temperature carbons, metallic carbides, and intermetallics. Tool steels show excellent wear properties, but have limited oxidation resistance. Unfortunately the carbons, metallic carbides, and intermetallics all present design and fabrication problems. As such, carbide coatings to be covered in the next section, offer the best approach to adapting these materials to piston ring and seal usage.

PISTON RING AND SEAL PLATINGS, COATINGS AND SURFACE TREATMENTS

A coating may be selected to combat any particular type of wear whether abrasive, adhesive, corrosive or a combination of these causes. Coatings also enable a manufacturer of wearing parts to select a base material which satisfies one requirement and a coating material which satisfies the wear requirement. For example, the base may be selected to withstand high temperatures; the coating in this instance would be selected to perform under high temperature wear.

Coatings for high temperature wearing parts, such as piston rings and seals, may be divided into three general categories: (a) wear resistant coatings, (b) break-in type coatings and (c) both wear resistant and break-in types. Five distinctly different methods are used to obtain these coatings as shown below:

I. Wear Resistant Coatings
   A. Hard metal deposits
   B. Carbide and oxide type coatings

II. Break-In Type Coatings
   A. Compositions containing solid lubricants
   B. Soft metal deposits

III. Both Wear Resistant and Break-In Types
   A. Hard or soft coatings created by thermal diffusion or chemical reaction with the base materials.

Table 3 shows a listing of coatings which have been grouped as Hard Metal Coatings, Carbide and Oxide Coatings, Solid Lubricant Coatings, and Coatings formed by Chemical Alteration of the base metal.

Each of these coatings have been classified according to Composition, Maximum Service Temperature °F, Material Compatibility Ratings, and Tested Values of Coefficient of Kinetic Friction. The test rig used to compile these data is shown in Fig. 6. Essentially this method of testing consists of applying the coating to the side face of a split piston ring and loading the coated face against a second ring of known composition and structure. The solid ring is rotated and the frictional drag between the mating surfaces is measured with a strain gauge linkage. The coated ring and the solid ring are weighed to the closest milligram on a chemical balance before and after testing, and the wear surfaces are carefully examined after testing for a surface compatibility rating. This rating is based

TABLE 3.—Piston Ring Coatings—Room Temperature Wear Ratings and Coefficients of Friction vs. Six Different Mating Surfaces

Coating Description	Max. Service Temperature Oxidizing Atmosphere (°F)	Wear Rating/Coefficient of Friction											
		T-15 Tool Steel		17-4 PH S.S.		Cast Iron BHN 201		Cast Iron BHN 432		Anodized Aluminum		Hard Cr. Plate	
HARD METAL COATINGS:													
Cr Electroplate	700	G	.30	P	.37	F	.42	F	.42	P	*	....	.....
Sprayed Mo	700	G	.52	P	.35	G	.55	G	.63	P	*	F	.45
Co-Ni Electroplate A	800	F	.40	P	.45	F	.42	F	.45	P	*	....	.....
Co-Ni Electroplate B	800	G	.45	P	.40	F	.34	F	.28	P	*	....	.....
Ni Chem. Plated	900	P	.27	P	.50	P	.46	P	.35	P	*	....	.....
Fe Electroplate	700	G	.47	P	.31	F	.40	G	.47	....	.....	G	.58
CARBIDE AND OXIDE COATINGS:													
85 Cr_3C_2—15 Ni-Cr	1300	G	.39	P	.30	G	.51	F	.57	P	*	....	.....
WC—7 Co	1000	G	.33	F	.35	G	.44	G	.36	P	*	....	.....
25 WC-7 Ni-68 WCrC	1200	G	.67	F	.64	G	.76	G	.64	P	*	....	.....
WC—12 Co	1000	G	.34	P	.40	G	.42	G	.33	....	.....	....	.....
80 WC—20 Binder	800	G	.35	P	.40	G	.65	G	.50	....	.....	P	.35
Al_2O_3	>1300	G	.51	F	.35	G	.47	G	.60	P	*	....	.....
SOLID LUBRICANT TYPE COATINGS:													
Graphite+MoS_2+Resin	800	G	.29	F	.18	G	.20	G	.20	F	.45	G	.29
W+MoS_2	600	G	.08	G	.06	G	.09	G	.08	G	.07	G	.11
MoS_2+Resin	600	G	.13	G	.14	G	.12	G	.14	G	.18	G	.15
MoS_2+Graphited Pb Powder	700	G	.14	G	.16	G	.15	G	.14	....	.....	G	.16
MoS_2 Formulation	800	G	.21	G	.11	G	.14	G	.15	....	.....	G	.18
SOFT METAL COATINGS:													
Ag Electroplate	600	G	.43	F	.70	F	.65	F	.65	P	*	....	.....
Sn Electroplate	600	F	*	F	*	P	*	F	*	....	.....	....	.....
Diffused Cd-Ni Electroplate	800	F	.45	F	.40	F	.32	F	.45	P	*	P	*
SURFACE TREATMENTS:													
Steam Trtmnt. (Cast Iron)	800	P	.35	F	.50	F	.56	F	.54	....	.....	P	.57
Malcolmized 17-4 PH	900	F	.35	P	.42	F	.55	F	.45	P	*	....	.....
Phosphated (Cast Iron)	600	....	.....	....	.....	G	.49	G	.58	....	.....	....	.....
Sulfurized (Cast Iron)	1000	....	.....	....	.....	F	.55	F	*	....	.....	....	.....
Nitrided Nitralloy 135M	1000	....	.....	....	.....	P	*	....	.....	....	.....	F	*

G = Good, F = Fair, P = Poor — *Not Obtained.

Fig. 6. Wear and Friction Test Rig.

on visual and low power binocular microscopic examination. The mating pair are then given an overall wear rating, based on weight loss and compatibility, described by one of three words: "Good", "Fair" or "Poor". For a pair to be rated "Good" they must possess relatively low wear weight loss and must show a smooth glazed surface. A "Fair" rating will be used if the weight loss is slightly higher and/or if the mating surfaces show indications of adhesive or abrasive wear. A "Poor" rating is used if high wear is encountered or if excessive adhesive or abrasive wear is observed.

The most promising of the coatings for elevated temperature service fall in the category of "Wear Resistant Coatings". The Hard Metal Deposits however, which include hard chromium plate, are not always considered as good candidates for elevated temperature wear. Chromium plate softens rapidly when subjected to heat, and the softening promotes adhesive type wear. For example, electro-chemically plated chromium in the as-plated condition possesses hardness on the order of Knoop 950 to 1100 (BHN 850-950). From the as-plated hardness, as stated, the hardness may be lowered to Knoop 550 (BHN 496) by heating to 1220 F.

Other hard metal coatings have not shown exceptional promise functionally, and possess an added disadvantage because of the difficulty of applying them to piston ring and sealing ring configurations. Application frequently results in altering the metallurgical structure of the base materials or involves excessive distortion of the coated part.

The Carbide and Oxide Coatings hold exceptional promise for elevated temperature piston ring and sealing ring applications. These materials, as a general rule, possess "Good" dry wear properties against mating surfaces of hardened tool steel and medium hardness and high hardness cast irons. This indicates that these materials resist welding, or if welding occurs it is of a covalent bond and is brittle and friable in nature (3). The metallic oxides and carbides possess metallurgical stability in the temperature range of 1000 to 1200 F and higher for certain coatings such as aluminum oxide. This is to say, they will retain their hardness which varies from DPH 1000 to 1450.

Solid Lubricant Type Coatings:

Practically all of the solid lubricant type coatings are formulations containing molybdenum disulphide as the primary constituent. Various liquid carriers are used which also serve as the bonding agents after the coatings have dried. Some of the higher temperature coatings of this type possess metallic additions such as tungsten. These additions plus a high temperature binder enable them to be used at higher temperatures. The authors' tests show, however, that molybdenum disulphide converts to a tri-oxide at temperatures on the order of 1000 F. The tri-oxide is very abrasive and this naturally is catastrophic to wear. As a general rule the molybdenum disulphide formulations are applied over a chemically etched base metal such as phosphating treatment used for cast iron and steel.

These coatings, for the most part, are classified as break-in type coatings. They are used for the purpose of establishing a compatible wearing surface; however, they cannot be expected to be permanent.

Coatings Formed by Chemical Alteration of the Base Metal:

Two methods presently being used for creating break-in type coatings by chemical conversion of the base metal are (a) steam treatment which creates an

TABLE 4.—Compatibility Ratings of Dry and Lubricated Brakeshoe Type Wear Tests

	LUBRICANTS								
RIDER MATERIAL[1]	DRY[2]	MLO 8200	F-50	81644	WRGL 31A	HEYDON NEWPORT	XF 258	81717	POLYESTER
Flame Plated WC	Good	Poor	Poor	Poor	Poor	Poor	Poor	Poor	Poor
T-15 Tool Steel	Good	Good	Good	Good	Good	Good	Good	Good	Good
Solid TiC	Fair	Fair	Good	Good	Fair	Good	Fair	Fair	Poor
T-4 Tool Steel	Good	Fair	.....	.....	.....	.....	.....	.....	.....
Cr-Ni-W-C0—"B"	Good	Fair	.....	.....	.....	.....	.....	.....	.....
Carbon "F"	Good	Good	.....	.....	.....	.....	.....	.....	.....
Carbon "G"	Good	Good	.....	.....	.....	.....	.....	.....	.....
Impregnated Carbon "D"	Good	Good	.....	.....	.....	.....	.....	.....	.....
Impregnated Carbon "E"	Good	Good	.....	.....	.....	.....	.....	.....	.....

[1]Ring Material—Flame Plated WC in all tests.
[2]Dry tests run 1 hour, lubricated tests 5 hours.

iron oxide coating (Fe_3O_4), and (b) immersion of ferrous parts in a manganese-iron phosphate solution which forms a black iron phosphide type coating. Both of these coatings are very thin, varying in thickness between .0001 in. to .001 in. depending upon the desired thickness. Their main advantage lies in the fact that they provide a certain amount of corrosion resistance, and the chemical conversion of the base material creates a matte-like surface which will hold oil or improve adhesion of the solid film lubricant type coatings.

Dry vs. Lubricated Wear Tests:

In the authors' testing programs for high temperature materials, as previously stated, the tests have been run on brake-shoe type wear test rigs without lubrication. This seemed like a logical and safe approach since in most cases the parts would be used at elevated temperatures where lubricants could not be used. Also it seemed like a safe assumption that if the wear properties were found to be good under conditions of no-lubrication, if a lubricant were used the wear properties should be ever so much better as a result of the lubricant.

One of the material combinations, Tungsten Carbide vs. Tungsten Carbide, which showed particularly promising results in room temperature and 1000 F temperature brake-shoe wear tests, was recommended to our experimental seal testing group. The application with which they were working was a 3 inch diameter rod seal unit to be used at moderately elevated temperatures (about 400 F) where a synthetic type lubricating oil was to be used. In this particular application the rod was Tungsten Carbide flame plated as was also the ID bearing surfaces of the rod seal. Much to our surprise, the Seal Testing Group soon reported a badly scuffed condition on the rod and seal bearing surfaces.

The Seal Testing Group then tried another material combination which had shown very successful dry wear properties. In this instance the materials combination was flame plated Tungsten Carbide vs. solid Titanium Carbide. Here again a scuffing condition was encountered although it took about ten times as long for the scuffing to occur as in the former case.

After these two reversals of bench type wear results were reported, it was decided to run further tests in which the mating pairs would be lubricated. Wear tests were repeated on the former dry combinations which again showed good results. These combinations were then tested in the same manner but under lubricated conditions. Eight different synthetic type lubricants were used. The results of these tests, shown in Table 4, corroborated the results found by the Seal Testing Group. The combination of Tungsten Carbide vs. Tungsten Carbide which showed "Good" results when tested dry showed "Poor" results in all instances where the testing was done under lubricated conditions. Also the material combination of Tungsten Carbide vs. solid Titanium Carbide was retested and revealed "Fair" wear compatibility dry, but was found to range from "Good" to "Poor" under lubricated conditions, depending upon the lubricant used.

The results of these tests suggest that wearing pairs which show "Good" results when tested dry and "Poor" results when tested under lubricants are creating a dry film lubricant which is a by-product of wear as they rub together in an atmosphere of air. When the air atmosphere is excluded, as it is when the lubricants are used, the dry film lubricant cannot form, and scuffing ensues as a result. Admittedly this is a theory based on few results, but we believe that further research should be done in this area to further understand the mechanism involved and to develop materials for high temperature wear based on this concept.

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A Study of the Torque Characteristics of GREASE-PACKED *R-2 and R-3* BALL BEARINGS

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The effect of grease type, grease amount, thrust load, and separator type is related to the torque characteristics of shielded R-2 (3.18mm bore) and R-3 (4.76mm bore) size ball bearings. Torque determinations were made at 1 to 2 rpm and at 15,000 rpm. Categorically:

1. Grease No. 1 (a diester oil thickened with a lithium soap meeting MIL-G-3278) yielded the lowest torque. Grease No. 2 (a synthetic oil lithium soap thickened grease conforming to MIL-G-7118) displayed the second lowest torque. Grease No. 3 (a silicone oil, aryl substituted urea thickened grease) resulted in the highest torque.

2. One piece cage configurations consistently yielded lower torque for the same load and grease type than did the two piece cages except in the case of R-3 bearings with grease No. 3, in which case torque was about equal for both cage types.

3. The torque level at 1 and 3 lb thrust loads was not much different for all greases and cage configurations tested.

INTRODUCTION

The purpose of this program was to study the interrelationships between grease type, separator configuration, grease content, and preload for use with R-2 and R-3 size bearings in order to yield optimum bearing torque characteristics. These bearing sizes were primarily designated for use in small electric motors.

The greases tested were:

(a) Grease No. 1, a diester oil thickened with lithium soap, meeting MIL-G-3278.

(b) Grease No. 2, a synthetic oil, lithium soap thickened grease meeting MIL-G-7118.

(c) Grease No. 3, a silicone oil, aryl substituted urea thickened grease.

Presented at 15th ASLE Annual Meeting, Cincinnati, Ohio, April, 1960.

TEST APPARATUS

A torque tester was constructed and is shown in Fig. 1. It consisted essentially of a vertically mounted shaft, the top end of which was shouldered in steps to accept the inner race of R-2, R-3, and R-4 size ball bearings. The shaft of the tester was mounted in ball bearings and driven by an air turbine. The test apparatus was capable of rotating speeds well in excess of 15,000 rpm, the top speed used in the bearing tests. In order to make a test the vertical driven shaft of the torque tester was fitted into the bore of the test bearing, and as previously mentioned, this vertical shaft was shouldered such that the face of the inner race of the test bearing seated on the shaft shoulder. A 1 lb. steel cylinder bored out to accept the face of the outer race of the test bearing was then placed over the test bearing, thereby applying a 1 lb thrust load on the bearing. This steel cup was hollow and could be filled with mercury. When filled the cup weighed 7 lb. The 3 lb thrust load tests were run with the cup filled with sufficient mercury to bring its total weight to 3 lb. A slotted shaft extended vertically from the top of the load cup, and accepted a blade that was connected to a spiral spring in such a manner that the spring supplied a resisting torque against the rotation of the blade. As the vertical drive shaft turned the inner race of the test bearing, the outer race was restrained from rotating by the counter torque of the spiral spring. The angular rotation of the blade attached to the spiral spring was calibrated in terms of torque on a dial, and thus provided the resisting torque indicating device. After suitable calibrations the torque test machine yielded repeatable and correlating results.

PROCEDURE FOR EACH GREASE TESTED

Twenty R-2 shielded bearings with 2 piece cages, and 20 with 1 piece cages were procured as well as 20 R-3 shielded bearings with 2 piece cages, and 20 with 1 piece cages. The shields were removed and the bearings identified by engraving. The test bearings were

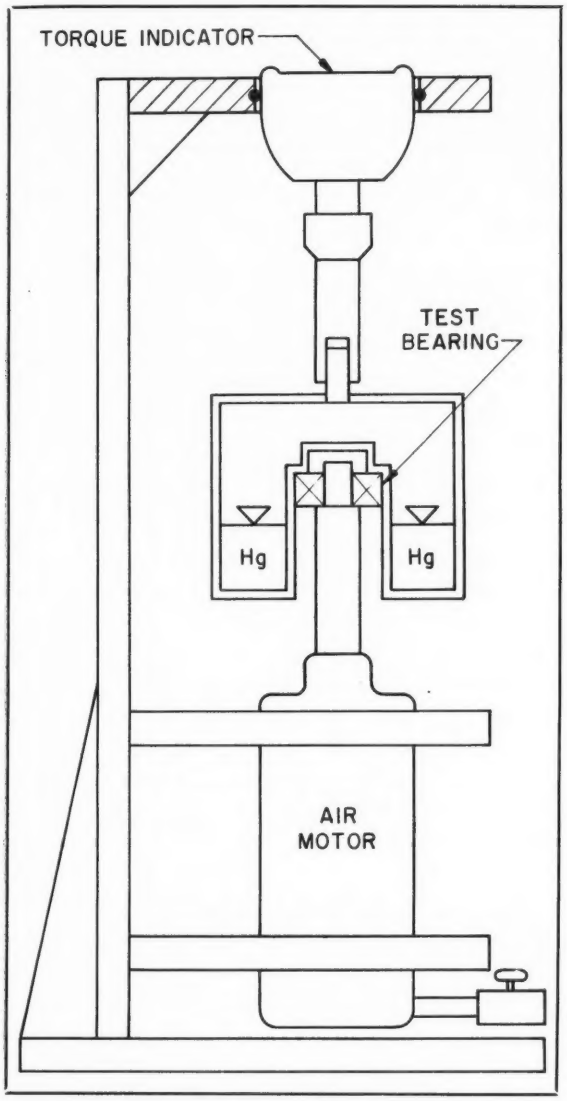

Fig. 1. Diagram of the torque testing apparatus used to obtain the relations between bearing torque and grease amount.

then demagnetized, cleaned, and lubricated with filtered instrument oil (1 drop for R-2 and 2 drops for R-3).

Torque tests were then performed on all bearings under 1 and 3 lb thrust loads at room ambient temperature. The torque at 1 to 2 rpm and at 15,000 rpm was obtained and repeated five times on each bearing, alternating the bearing faces, in order to check reproducibility, and data scatter, and to secure as representative a torque determination as possible and still be economical of time. The test bearings were then cleaned and greased, utilizing an analytical balance, with the grease weights shown in Table 1.

The test bearings were then torque tested at 1 to 2 rpm and at 15,000 rpm under 1 and 3 lb thrust loads in a manner similar to the testing done when they were lubricated with instrument oil.

TABLE 1.—Bearing Grease Weight Schedule

R-2 bearings			R-3 bearings		
Cage type	Number of bearings greased	Weight of grease (mg)	Cage type	Number of bearings greased	Weight of grease (mg)
2 pc	5	4±1	2 pc	5	15±1
1 pc	5	4±1	1 pc	5	15±1
2 pc	5	8±2	2 pc	5	30±2
1 pc	5	8±2	1 pc	5	30±2
2 pc	5	14±2	2 pc	5	40±2
1 pc	5	14±2	1 pc	5	40±2
2 pc	5	20±2	2 pc	5	50±2
1 pc	5	20±2	1 pc	5	50±2

CONCLUSIONS AND DISCUSSION OF RESULTS

Figs. 2 and 3 display the results of the tests previously described. The curves represent torque vs weight of grease at 1 to 2 rpm, and at 15,000 rpm. The individual graph headings are self-explanatory as to grease type, thrust load, bearing configuration, etc. Five bearings were tested for each grease weight, and each of these five separate bearings were tested five times, this means that each plotted point represents the average of 25 separate torque determinations for a given grease weight.

The cross-hatched areas surrounding each curve represents the approximate torque spread one might expect for a given grease weight.

The torque at zero grease weight is the average torque at 1 to 2 rpm or at 15,000 rpm due to two drops of instrument oil in the case of R-3 bearings and one drop of instrument oil in the case of R-2 bearings. This zero grease weight torque is the average instrument oil torque of all bearings tested of a given configuration—for instance, the zero grease weight torque for all R-2, one piece cage bearings under 1 lb thrust load, is the average oil torque of all these bearings (20 tested). This torque was determined before the bearings were greased and with no shields on the bearings.

One of the assumptions underlying this test program is that the shields on the bearings were not rubbing and did not affect the torque in any manner, regardless of shield configuration. All shields were installed while viewing the operation under 25 power magnification, and all shields were similarly inspected for damage or rubbing before testing. No evidence of shield rubbing was observed upon inspection before or after testing.

Once again, referring to the graphs, 15,000 rpm and 1 to 2 rpm torque curves are designated by appropriate notes. Straight dashed lines connecting the zero grease weight torque point to the lowest grease weight torque point on some graphs indicates that there is not enough information to justify a curve, and does not indicate that the torque is believed to vary as shown between these two points. The results do display considerable scatter in some instances, but this scatter is not believed to be excessive or unexpected due to the statistical character of grease-packed ball bearings.

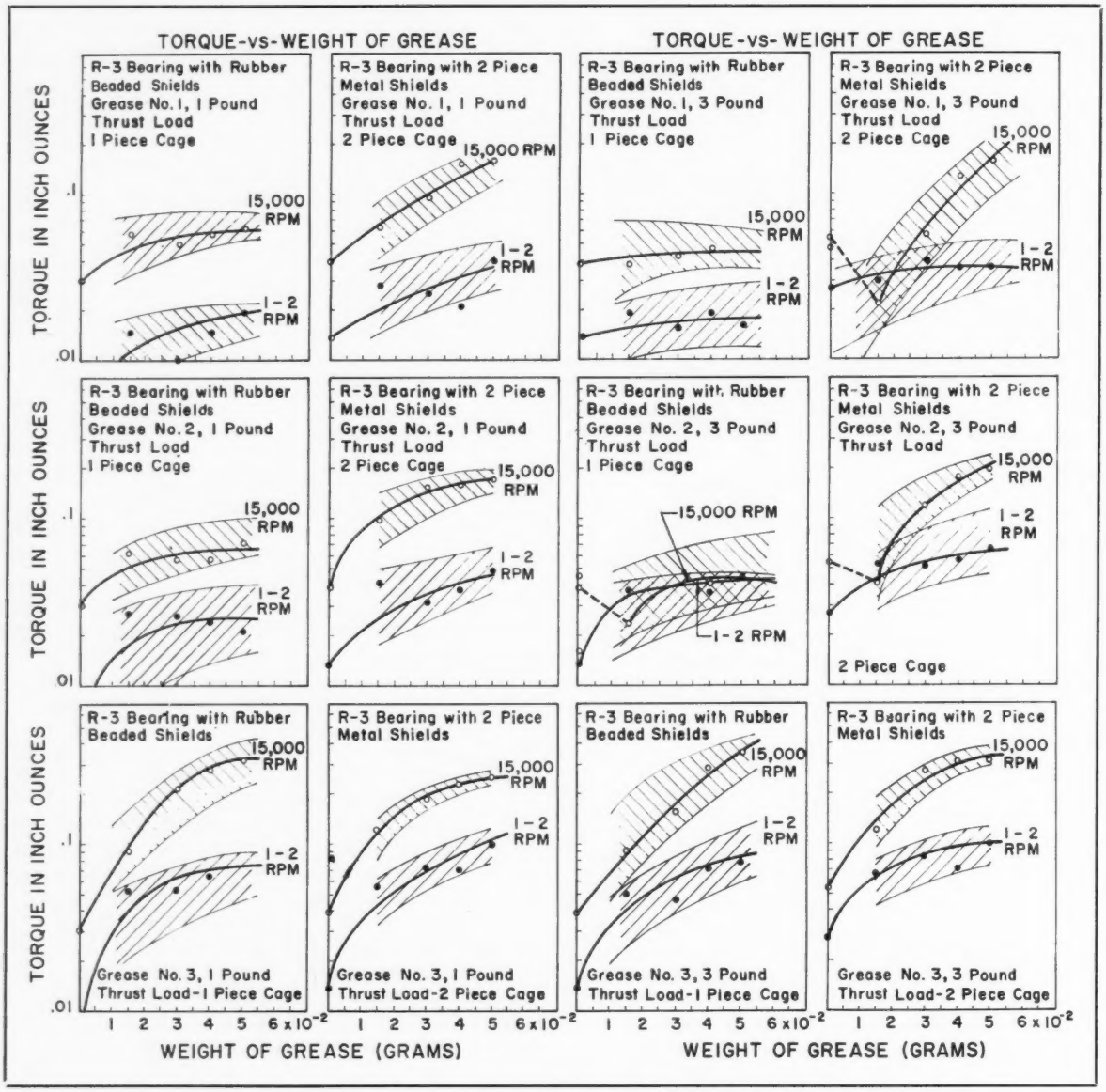
TORQUE-vs-WEIGHT OF GREASE
TORQUE-vs-WEIGHT OF GREASE
TORQUE IN INCH OUNCES
R-3 Bearing with Rubber Beaded Shields Grease No. 1, 1 Pound Thrust Load 1 Piece Cage
R-3 Bearing with 2 Piece Metal Shields Grease No. 1, 1 Pound Thrust Load 2 Piece Cage
R-3 Bearing with Rubber Beaded Shields Grease No. 1, 3 Pound Thrust Load 1 Piece Cage
R-3 Bearing with 2 Piece Metal Shields Grease No. 1, 3 Pound Thrust Load 2 Piece Cage
R-3 Bearing with Rubber Beaded Shields Grease No. 2, 1 Pound Thrust Load 1 Piece Cage
R-3 Bearing with 2 Piece Metal Shields Grease No. 2, 1 Pound Thrust Load 2 Piece Cage
R-3 Bearing with Rubber Beaded Shields Grease No. 2, 3 Pound Thrust Load 1 Piece Cage
R-3 Bearing with 2 Piece Metal Shields Grease No. 2, 3 Pound Thrust Load
2 Piece Cage
R-3 Bearing with Rubber Beaded Shields
R-3 Bearing with 2 Piece Metal Shields
Grease No. 3, 1 Pound Thrust Load-1 Piece Cage
Grease No. 3, 1 Pound Thrust Load-2 Piece Cage
Grease No. 3, 3 Pound Thrust Load-1 Piece Cage
Grease No. 3, 3 Pound Thrust Load-2 Piece Cage
15,000 RPM
1-2 RPM
.1
.01
1 2 3 4 5 6 x10⁻²
WEIGHT OF GREASE (GRAMS)

Fig. 2. Comparison of torque characteristics of R-3 bearings under thrust load.

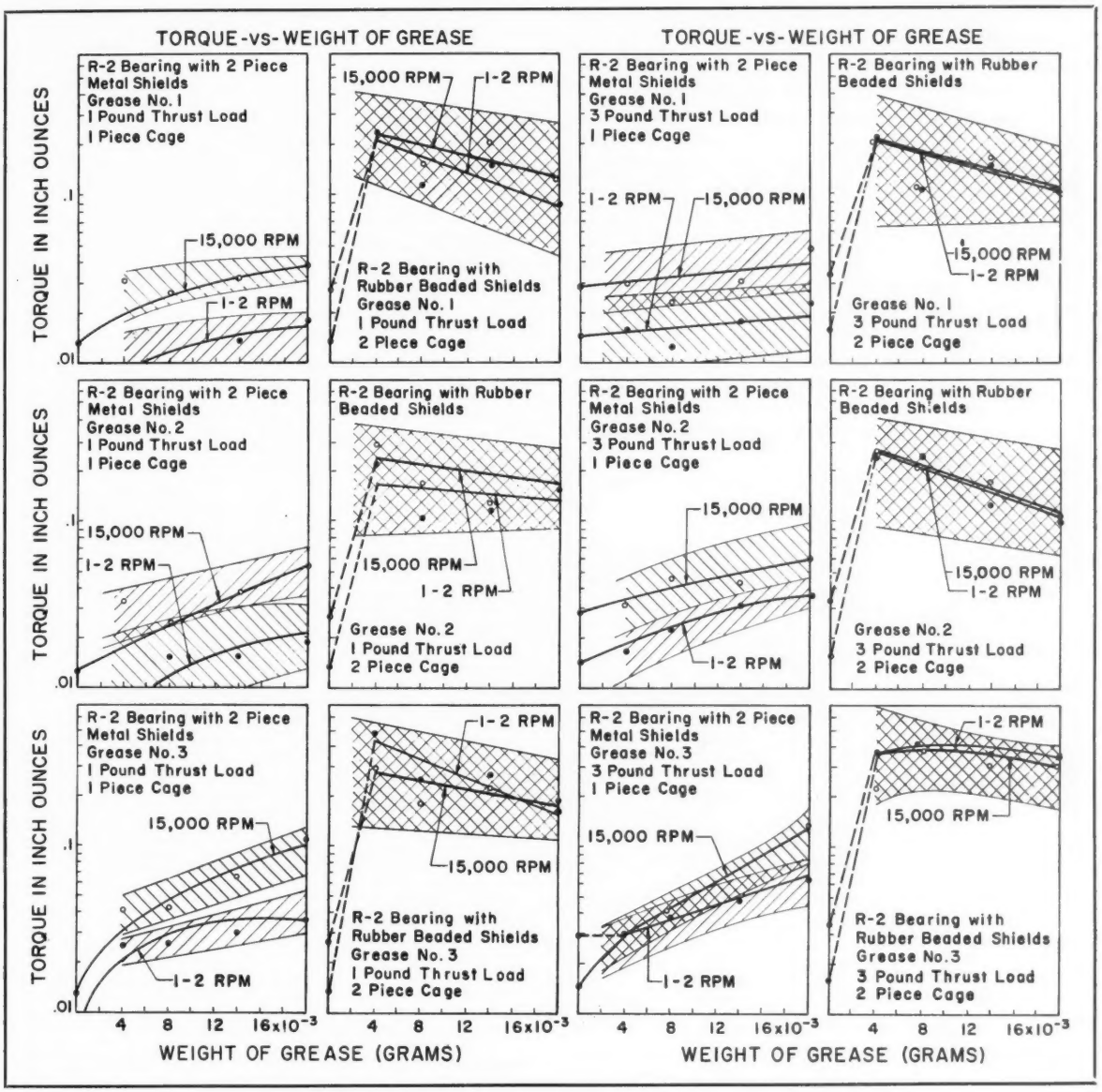
TORQUE-vs-WEIGHT OF GREASE
TORQUE-vs-WEIGHT OF GREASE
TORQUE IN INCH OUNCES
R-2 Bearing with 2 Piece Metal Shields Grease No. 1 1 Pound Thrust Load 1 Piece Cage
15,000 RPM
1-2 RPM
R-2 Bearing with Rubber Beaded Shields Grease No. 1 1 Pound Thrust Load 2 Piece Cage
R-2 Bearing with 2 Piece Metal Shields Grease No. 1 3 Pound Thrust Load 1 Piece Cage
R-2 Bearing with Rubber Beaded Shields Grease No. 1 3 Pound Thrust Load 2 Piece Cage
R-2 Bearing with 2 Piece Metal Shields Grease No. 2 1 Pound Thrust Load 1 Piece Cage
R-2 Bearing with Rubber Beaded Shields Grease No. 2 1 Pound Thrust Load 2 Piece Cage
R-2 Bearing with 2 Piece Metal Shields Grease No. 2 3 Pound Thrust Load 1 Piece Cage
R-2 Bearing with Rubber Beaded Shields Grease No. 2 3 Pound Thrust Load 2 Piece Cage
R-2 Bearing with 2 Piece Metal Shields Grease No. 3 1 Pound Thrust Load 1 Piece Cage
R-2 Bearing with Rubber Beaded Shields Grease No. 3 1 Pound Thrust Load 2 Piece Cage
R-2 Bearing with 2 Piece Metal Shields Grease No. 3 3 Pound Thrust Load 1 Piece Cage
R-2 Bearing with Rubber Beaded Shields Grease No. 3 3 Pound Thrust Load 2 Piece Cage
.1
.01
4 8 12 16x10-3
WEIGHT OF GREASE (GRAMS)
WEIGHT OF GREASE (GRAMS)

Fig. 3. Comparison of torque characteristics of R-2 bearings under thrust load.

The results the data indicate can be summarized as follows:

1. Grease No. 1 consistently yielded the lowest torque for all bearing and load configurations tested. Grease No. 2 displayed second lowest torque readings, and Grease No. 3 resulted in the highest torque of the three greases tested.

2. One piece cage configurations consistently yielded lower torque for the same load and grease type than did the two piece cages except in the case of R-3 bearings with Grease No. 3, in which case torque was about equal for both cage types.

3. The torque level due to 1 lb and 3 lb thrust loads for all greases and cage configurations can be obtained directly from the enclosed graphs. In general, the torque level is not much different.

4. The R-2 bearings with the two piece cage configuration under 1 lb and 3 lb thrust loads displayed the odd property of decrease in torque with increasing grease content for all greases. This may be because in the region tested the major retarding torque is due to cage drag and the larger grease contents supply better lubrication to the 2 piece cage than do the smaller grease amounts.

ACKNOWLEDGEMENTS

This work was done while the author was employed by AiResearch Manufacturing Company of Los Angeles, a member of the Garrett Corporation and was carried out in their research facilities. Appreciation is expressed to them for graciously allowing the data to be published.

The purposes of this paper are (a) to present information necessary for design of lubrication systems, and (b) to show where aluminum might be used to great advantage to the people who are most interested in its use — the production and maintenance personnel.

Designs of systems are usually made by engineers; either lubrication, mechanical or chemical. Some designs are provided by maintenance department personnel and some by salesmen or sales engineers. It is believed that satisfactory designs can be provided quickly when the designer has a satisfactory set of specifications. Of course, he must also know where such specifications are applicable.

Aluminum alloy piping and fittings, etc., can be used advantageously in many systems. The advantages of its use are given in the paper along with comparative costs. Limitations of applicability are also shown.

by Ewell E. McDole
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Use of Aluminum in LUBRICATION SYSTEMS *—Design and Specifications*

INTRODUCTION

The purpose of this paper is to present considerations of design for lubrication systems highlighting design and utilization of aluminum with the attendant advantages and disadvantages.

Included are specifications for aluminum piping and plate and for pickling of steel pipe. These specifications were used in construction of lubrication systems for several mills of a large rolling mill installation.

DISCUSSION

In general, a lubrication system consists of the following components: tank reservoir, piping, pumps, filters or strainers, air-cushion or surge tank, cooler, temperature and pressure controls, and an alarm system.

For the purpose of this paper, primary attention will be paid to tank or reservoir, cooler and piping.

The first step in the design of a lubrication system might be the determination of required tank or reservoir capacity. Capacity is based on lube oil flow rate and viscosity with consideration given to materials to be settled or floated. Lube oil flow rate is generally available as a part of machine design. Where flow rate is not available, good design criteria can be obtained from Kirschner (1).

Operating volume of the tank is nearly always 10 to 40 times the flow per minute. The smaller volume is generally satisfactory where there is time required only for breaking of foam and the larger volume is used where settling of water or dirt is required from viscous oils. The actual tank volume will, of course, be the operating volume plus drain-back when the system is shut down. The volume of drain-back will rarely be more than the volume of the return lines, but the level buildup from individual gear cases and bearings under operating conditions must be estimated in the establishment of size.

Tank dimensions are preferably width=1 to 1.5 height and length=2 to 12 height at operating level. Within these limits, optimum settling with respect to tank cost is obtained.

Major factors of tank design are heat dissipation, tank support, baffles, inlet and outlet connections, heating or cooling, drainage, tank cover and access hatches or cleanout openings. Fittings may be desired such as for wells of temperature controller bulb or bulbs, ther-

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mometer, gage cocks, floats or gage glass fittings, breather, fill line, and possibly, a place for tank level alarm.

Heat dissipation requires, practically, that there be air flowing by the tank. The amount of heat dissipated from any tank (and piping) can be estimated from information found in handbooks. The heat gain can be estimated by assuming inefficiency of gears and bearings to be converted to heat (42.4 BTU equals 1 HP minute). By trial and error, the approximate operating temperature can be obtained. If this is too high, say above 130 F in the summertime, a cooler should be supplied.

Coolers are readily available made of aluminum, steel, copper and brass, in either tube bundle or finned-tube types. There are many applications where air-cooled, finned-tube exchangers are preferred because of initial installation cost or by absence of danger of contamination from water leakage.

Tank supports, either tanks on legs or set on a concrete pad, are a minor cost consideration. Tanks on legs favor maximum heat dissipation, good tank drainage and permission for detection and repair of leaks with a minimum of corrosion on the bottom.

Baffles for obtaining uniform flow may serve also as reinforcing for tank walls and top. In small tanks, there may be only one baffle needed (it may be a simple plate or screen), and for larger tanks, baffles might be as shown in Fig. 1.

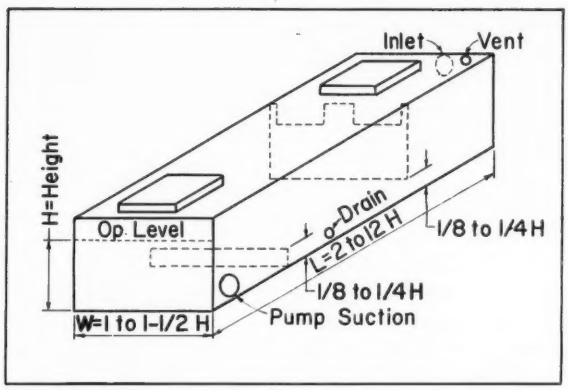

Fig. 1. General tank design.

Inlet and outlet flanges usually need to be as close to the tank as possible to minimize room required for piping. This often calls for tapped holes in, but not through, a special flange made to set in the tank wall.

Heating coils inside tanks should be as close to the bottom as practical to permit cleaning of tank bottom. There should be a slight slope in the coils—at least 1/16 of an inch per foot of pipe—to minimize "steam-hammer." The coil area should be selected for a low coefficient of heat transfer. The satisfactory rate to permit required heatup when fouled is 50 to 60 BTU per square foot per degrees F for plain pipe and a correspondingly low figure for finned pipe. ASA Schedule 80 seamless steel pipe, ASTM Grade A-53, is preferred for heating coils inside the tanks. Low pressure steam of 35 PSI or less is usually preferred for heating lube oils. Electric heaters can also be used. These, however, must be a very low watt-density type to prevent sludge formation with many oils. Cost of electrical heaters, wiring and controls is usually several times the cost of steam or hot water heating.

The tank top serves to hold an inspection port or access hatch to each of the compartments, vent opening or openings, and, possibly, lube return pipe, level alarm and float sheave support. There are times when lifting lugs for the empty tank might be provided on the tank top.

When a tank breathes air saturated with moisture above the dew-point of the oil temperature or tank top, water will condense. It is highly preferable to keep moist air out. If, however, moisture enters, a constant stream of dry air through the breather may carry away enough water to prevent need for settling and/or centrifuging. Reference to psychometric charts will permit estimation of air required to keep oil dry.

Normally, the breather should be a separate tank opening to prevent an eductor type of situation from drawing in moist air when the breather is a part of the return oil inlet. Breather air in most applications should be filtered and/or dried.

Factors discussed for tank design are for either iron or aluminum construction, but primary attention is focused now on the utilization of aluminum in lubrication systems.

Once acquainted with the advantages of aluminum in lubrication systems, it is expected that it will be used wherever possible. As all other metals, however, aluminum has both advantages and disadvantages. It might be well to recognize, on the basis of past experience, just what are these advantages and disadvantages.

All common mineral oils, multipurpose extreme pressure oils, and synthetic hydraulic fluids are satisfactory for use with aluminum tanks, piping and valves when the systems are kept dry. The principal advantages of an aluminum tank are the assurance that paint will never peel off the inside to cause trouble or down-time (since there is no need for paint), low weight, and no catalytic oxidation effects on lube oils. Aluminum piping is more rapidly fabricated and can be absolutely cleaned by flushing. There is never danger to bearings or gears from rust or scale. Aluminum tubing and piping is expected to be satisfactory in normal water environment—in the range between pH 6 and 8.5.

It is expected that leaded gear oils would *not* be satisfactory in the presence of water, for it is known that in the presence of water, a lead-aluminum couple results in pitting of aluminum. For this reason, the use of aluminum cannot be recommended in this important application at this time. Where there is question of satisfactory application, aluminum alloys, like all other materials, should be tested in the environment where aluminum might be used.

In order to determine whether or not aluminum alloys are practical and/or desirable in the general area of tank construction, some comparative data is necessary. The data that follows is based on recent installations.

TANK CONSTRUCTION

Use of alloys 5052, 5454, 5456 or 5083, Aluminum Association designations, is usually recommended. These are all weldable with little loss in strength. The temper can be selected from *Aluminum Construction Manual*, which is available from The Aluminum Association, 420 Lexington Avenue, New York, New York.

Weld filler material used is according to Standard Specification No. 2.

TABLE 1.—COST COMPARISON OF A 4 x 4 x 8 FT RECTANGULAR TANK

	ALUMINUM @ 50c/LB		STEEL @ 13c/LB	
Plate	¼″	$340	3/16″	$187
Welding		144		110
Sandblasting		—		60
Prime and Paint		—		60
Contractor Fee, 0.4 x Cost		193		167
TOTAL		$677		$584

Cost of the aluminum tank is 116% of; that is, 16% above, the cost of the steel tank. When fittings, hatches, etc., are added to the tank, the total cost of the aluminum tank will be about 118 to 120% of the cost for a steel tank. By the time the tank is shipped, unloaded, and set in place, the cost will be about 110% of a steel tank—with its first coat of paint. Costs of smaller tanks below about 300 gallons will often favor the use of aluminum, cost-wise. This should be of much interest to machine shop equipment manufacturers and users.

In discussing piping design, factors of important differences from steel piping will be stressed. Functionally, the main advantages in aluminum for lube oil piping are that pickling is not required, the flushing process goes faster, and it will not rust or corrode after flushing. The piping system is ready for use almost as soon as it is tested. After complete flushing, there can be no dirt or scale to damage gears or bearings—provided there are adequate settling tank, screens and/or filters to remove wear debris. The size of aluminum piping recommended is the same as would be used for other smooth-wall materials.

Aluminum piping can be either threaded or welded —or a combination. Threads should be lubricated and sealed with a suitable plastic tape, such as polytetrafluoroethylene 3 or 4 thousandths inch thick, ½ to ¾ inch wide.

Aluminum walls of Schedule 10 pipe are perfectly acceptable for welded piping when it is not subject to physical damage. When using Schedule 10, flanges of the same inside diameter must be used in order to permit obtaining a clean system by flushing. Piping in Schedule 10, 20 and 40 is all readily available. Wall thicknesses in Schedule 20 and 40 aluminum pipe are the same as for ASA Schedule 20 and 40 for wrought iron pipe, Standard B-36.10. Thicknesses for aluminum pipe in Schedule 5 and 10 are those used for stainless steel of the ASA Schedules.

One major difference between aluminum and steel is the coefficient of expansion. That of aluminum alloys is about four times as great as for steel. There will be times when an expansion joint will be required with aluminum and not with steel. A bellows-type or tire-type of stainless steel or synthetic rubber, respectively, will usually be satisfactory and add little to cost.

In lube systems where water is not present, either inside or outside, there will be no harm in mating aluminum and steel, cast iron or brass. When walls will be wet with water, an insulating joint must be used. This joint is very simple in that it consists of a synthetic rubber or plastic gasket ⅛ inch thick between flanges joined by stainless steel, cadmium or zinc-coated bolts.

Specification No. 1 covers material requirements for aluminum piping in detail. Specification No. 2 covers filler materials for welding of aluminum used in piping and in tanks. These specifications, as well as specifications for demillscaling of steel piping (Specifications No. 3 and 4), are included.

Aluminum piping use in lubrication systems is relatively new to the writer and his company. The following comments of comparative costs are based on four systems of welded pipe and one system of threaded pipe—all installed by construction workers.

Cost comparisons for welded piping need to be divided into two parts based on pipe size. Cost of installation is slightly (1 to 10%) lower than steel, all Schedule 40, for sizes larger than 4 inches. Material costs for an all-aluminum system will be about four times the cost of steel pipe and fittings and cast iron valves. This will result in an aluminum lube piping job ready for use at about 95 to 110% of the cost of a steel piping system ready for use after being demillscaled according to Specification No. 3.

The cost of welded aluminum piping installations below 4 inches, again Schedule 40, are a little higher relatively than the larger sizes. Here, the costs can be expected to be as much as 150% of steel piping costs for small pipe sizes at the consumption and availability of aluminum fittings in the latter part of 1959.

Ease and desirability of installation of welded aluminum piping can be shown by evaluation of general foremen for contractors. These foremen supervised the installation of piping on two mills. Both feel that (a) contractor profit margin is higher with aluminum, (b) aluminum is prepared more quickly for welding and welding is much faster on aluminum, (c) when modifications are necessary, there is no danger of having to re-pickle the system, and (d) no need for paint or painters.

Certified welders for aluminum piping in the Mid-Ohio Valley area amount to about 40% of the piping welders in the construction forces. Many of these have been trained on construction jobs. Those trained, prefer welding aluminum to any metal they have welded, according to unsolicited comments. Welders training manuals are available from suppliers of aluminum through their sales representatives.

A general view of piping is given in Fig. 2 of a lubrication system for a mill equipped with sleeve bearings.

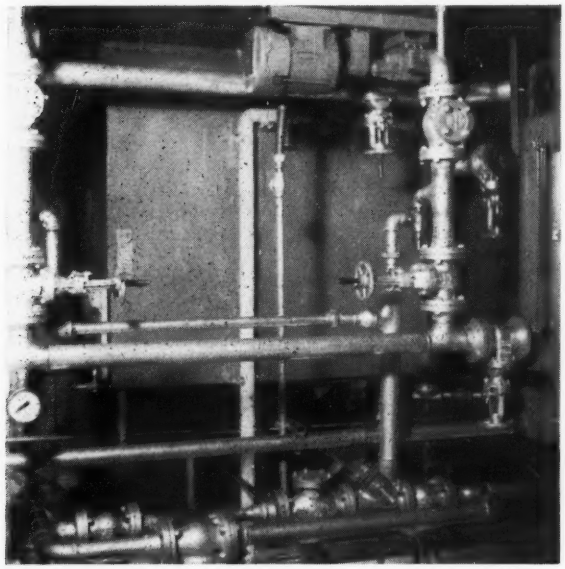

Fig. 2. Lubrication system piping for a mill equipped with sleeve bearings.

REFERENCES

1. Kirschner, J. B., "Specification of Circulating Oil Systems," *Lubrication Engineering*, Vol. 15, No. 12—1959 pp. 484, 485, and 496.

SPECIFICATION NO. 1

Aluminum Lube Oil Piping — 125 psi

NOTE: These Specifications shall become effective only when reference to this Specification number is made on working drawings. In case of conflict between working drawings and referenced specifications, the working drawings shall govern.

Pipe

⅛ in. to 1½ in.—3003-H112 or 6063-T6 extruded ASA Schedule 40 pipe.
2 in. to 6 in. —6063-T6 extruded ASA Schedule 40 pipe.
8 in to 12 in. —3063-T6 extruded ASA Schedule 30 pipe or 3003-H14 roll formed and welded Schedule 30 pipe.

Piping Joints

⅛ in. to 1½ in—Threaded and coupled.
2 in. to 12 in. —Welded and flanged.

Fittings

⅛ in. to 1½ in.—Standard B214 aluminum alloy thread fittings.
2 in. to 12 in. —3003-F or 6061-T6 standard weight welded fittings for butt welding.

Flanges

½ in. to 1½ in.—ASA Standard 125 lb screwed.
2 in. to 12 in. —ASA Standard 125 lb slipon or welding neck.

Gate or Ball Valves

¼ in. x ⅜ in. —Aluminum 100 lb WP, inside screw rising stem, screwed body, integral seats.
½ in. to 1½ in.—Aluminum 100 lb WP, inside screw rising stem, screwed body, stainless steel seats, stems, and wedges.
2 in. to 8 in. —Aluminum 100 lb WP, outside screw and yoke, flanged, stainless steel seats, stems and wedges.

Globe and Angle Valves

¼ in. to 1½ in.—Aluminum 100 lb WP, inside screw rising stem, screwed body, stainless steel seats and stems, composition disc.
2 in. to 3 in. —Aluminum 100 lb WP, inside screw rising stem, flanged, stainless steel seats and stems, composition disc.

Check Valves

¼ in. to 1½ in.—Aluminum 100 lb WP, swing gate, screwed body, stainless steel hinge pin.
2 to 3 in. —Aluminum 100 lb WP, swing gate, flanged, stainless steel hinge pin.
4 in. to 12 in. —Cast Iron 125 lb Body, swing gate, flanged, all iron fittings.

Plug Valves

½ in. to 1½ in.—Aluminum 100 lb WP, screwed body, wrench operated.

Bolting

All sizes—Galvanized or cadmium plated square head machine bolts and heavy hex nuts conforming to ASTM A-307, Grade B.

Gaskets

All sizes—1/16 in. thick neoprene, full face for service temperatures below 200 F, asbestos composition for service temperatures above 200 F.

Pipe Supports

Pipe supports shall be of conventional design, except that all elements in contact with aluminum shall be painted with primer and one coat of alminum paint.

Protective Coating

None required.

SPECIFICATION NO. 2

Filler Material for Welding Aluminum by the Inert Gas Tungsten Arc Process

Scope

This Specification covers the filler materials to be used for welding various aluminum alloys.

TABLE 2.—Filler Material for Welding of Aluminum by the Inert Gas Tungsten Arc Process

BASE METAL	FILLER MATERIAL
1100 3003	1100 or 4043
3004 5005 5050 6061 6062 6063	5356 or 4043
5052 5154 5083 5086	5183 or 5356

SPECIFICATION NO. 3

Descaling Hydraulic and Lube Oil Piping Systems

Scope

Scope—The work under this section consists of furnishing all labor, equipment and material necessary to completely clean, pickle, neutralize, passivate an oil hydraulic and lube oil piping system as specified herein.

Definition—The term system, as used herein, shall mean all the piping on any separate hydraulic or lube oil system of any individual mill, machine or equipment line.

General Requirements

Coordination of Work—The work to be performed hereunder shall be coordinated with that of other construction parties. Arrangements shall be made to furnish or provide air, water, and steam supply lines and connections to the various hydraulic and lube oil systems which are to be treated well in advance of the beginning of the pickling process. Arrangements for hauling away and properly disposing of the degreasing, pickling and oil solutions and for discharging other solutions into storm sewers shall also be made.

Materials and Equipment—All necessary chemicals, materials, and equipment shall be supplied.

Enough pumps, hose and hose connections, tanks, heating facilities and hammering equipment shall be provided to permit processing of two systems simultaneously. For example, it shall be possible to wash, degrease, rinse, and pickle one system while passivation and oil treating is being done on another system. Tank pickling of the short lengths of pipe and fittings which have been removed is being pickled by the circulation method.

Pumps and supply and return lines shall be capable of providing and maintaining circulation of all solutions at the rate of 7 to 10 feet per second through all parts of each system.

All hose used shall be oil treated.

Air Hammers—Small air hammers with saddles shall be provided for striking pipe as circulation progresses.

Joint Hammering—One pound hammers shall be provided for hand-hammering joints during circulation.

Heating Facilities shall be capable of heating solutions to and maintaining them at the temperatures specified herein.

Inspection of Setup—Before starting the pickling process of any hydraulic or lube oil system, the entire system shall be carefully inspected to make certain that all parts which might be damaged by scale, rust, or acid have been isolated or removed from the system.

After the circulating pumps, tanks and lines have been connected to the hydraulic or lube oil system, the entire setup shall be inspected for the following:

a. Water, steam, air, pumping, and drain connections.
b. Interconnections from supply to return line or lines.
c. Pump capacities and line sizes.
d. Ample heating facilities.

General Procedure

Leak Test—After a complete visual inspection, water shall be circulated through the system and discharged into the sewer. Joints shall be hammered during circulation and all connections shall be checked for leaks. If leaks occur, lines shall be emptied, repairs made, and the system rechecked by circulating water again. The system shall be entirely leak-proof before proceeding with the descaling processes.

Degreasing—The hydraulic or lube oil system shall be connected to the supply and return lines of a tank suitable for circulating and heating the alkaline cleaner. The cleaning solution shall contain phosphates, carbonates and silicates suitable for removing all oils and greases. The degreasing solution shall be heated to and maintained at 125 to 135 F while being circulated through the system. Circulation period shall last at least two hours and shall continue as long thereafter as necessary to remove all oil and grease from the lines of the system. All joints shall be hammered during the circulation period. The degreasing solution shall not be allowed to drain into the sewer. It shall be dumped into a ground pit.

Water Rinse—After all degreasing solution has been drained from the system, water shall be circulated through the system and drained into the storm sewer. The rinsing shall continue for 30 minutes. Care shall be taken to avoid thermal shock to the piping system when admitting cold rinse water.

Pickling

Pickling Solution—The pickling solution shall be mixed in the tank which will be used to supply the system during the pickling process. A strainer shall be installed at the end of the discharge line through which the pickling solution returns to the tank. This strainer shall be cheesecloth or one of the acid-proof fibers of approximately the same weave as cheesecloth. If a cheesecloth strained is used, it shall be replaced every two hours.

The pickling solution shall be prepared as follows:

a. An appropriate amount of inhibitor must be added to each carboy of 20°Be. muriatic acid to prevent attack of metal.

b. The mixture of muriatic acid and inhibitor shall be diluted with 80 gallons of water by adding the acid to the water.

c. A wetting agent shall be added to the diluted solution. The amount of wetting agent shall be according to manufacturer's recommendations.

Pickling Procedure—The pickling solution shall be started through the system at room temperature and then shall be heated to and held at a temperature of 160 plus or minus 5 F throughout the entire circulation period. Piping and joints shall be hammered continuously during the 6 to 8 hours of pickling.

Inspection—Circulation shall continue until inspection of parts least likely to be demillscaled show the scale to be completely removed. Inspection is done by interrupting circulation, displacement of acid with nitrogen in the area to be checked, disassembly, visual checks, re-assembly, etc.

Displacing the Acid—When the pickling is completed, the acid shall be displaced from the entire system by nitrogen. Pumps shall be emptied and rinsed.

Note: The pickling solution shall be neutralized and hauled to a pit where it will be allowed to seep away or it may be dumped into a pit and neutralized there. In order to neutralize 100 gallons of pickling solution, 66 pounds of soda ash or 44 pounds of flake caustic soda are required. Either of these alkalies must be in solution before being added to the pickling solution.

Water Rinse—As soon as the pickling solution is removed from the system and the pumps, they shall be flushed with circulating water for approximately 30 minutes. This water shall be discharged into the storm sewer. At the end of this rinse period, the system shall not be drained, but the solution of trisodium phosphate and sodium *nitrite* (passivating solution) shall be started through the system immediately. (All drain points must be flushed free of acid during this period.)

Passivation—A tank of warm, trisodium phosphate solution containing 20 pounds TSP and 10 pounds of sodium *nitrite* per 100 gallons of water shall be ready to be pumped through the system as soon as the water rinse is completed. Outflow change from sewer to circulating tank shall be made as soon as the phosphate solution shows up at the outlet valve. This solution shall be heated while circulating until the temperature of the solution as it returns to the tank has reached 170-200 F. Circulation of this solution through the system will form a rust-resistant coating on the fresh iron surfaces. Upon completion of this process, the solution may either be discharged into the storm sewer or hauled to the pit.

Draining—The system shall be allowed to drain freely for one hour.

Oiling—After the draining is completed, a rust-inhibiting flushing oil of about 100 SSU at a temperature of 100 F shall be started through the system. While circulating, the temperature of the oil shall be raised to and maintained at approximately 170 F. (If the oil contains water, it shall be heated to boiling.) Circulation of the hot oil shall continue for at least 24 hours. The entire system shall be hammered from one end to the other until no scale is removed in a two-hour hammering period. The cheesecloth or fiber strainer shall be used to trap scale or dirt as the oil returns to the tank. When, after two hours of hammering, no scale or dirt is caught in the screen, and the oil has circulated at least 24 hours, the system shall be completely drained of oil. No oil shall be discharged into the sewer.

SPECIFICATION NO. 4

Demillscaling Small Sections of Piping

Scope

This specification covers the demillscaling procedure for small sections of ferrous piping where circulation of the system is not possible or required, and where the piping will be filled with oil immediately following the demillscaling operation. This specification is not intended to apply to descaling of complete piping systems where circulation is required during descaling.

General Requirements

The following procedure shall be used for the work covered by this specification:

a. All sections of piping shall be thoroughly degreased with cold solvent.

b. Degreased sections shall be thoroughly cleaned of grease solvent with live steam.

c. After steam cleaning, all joints in piping sections shall be hammered to loosen scale, using a one-pound hammer.

d. Piping sections shall be held in a vertical position, lower openings capped or plugged, and piping sections completely filled with an inhibited acid approved by the Owner. Piping sections shall be arranged during descaling to prevent pockets of either air or evolved gas. Descaling solution will be allowed to stand in pipe sections for a minimum of 24 hours and a maximum of 30 hours.

e. After descaling as described above, the descaling solution will be emptied into open drums and the piping section will be immediately rinsed with sanitary water. After rinsing, the section shall be thoroughly heated with live steam and then blown completely dry with compressed air. After drying with air, the interior surfaces of the section shall be completely oiled with a suitable oil approved by the Owner.

f. The Contractor shall make provision to keep all descaled piping sections completely clean after drying and oiling and until installation is complete.

g. If, after the descaling process, it becomes necessary to weld any piping sections which have been descaled, that piping section shall be removed and the descaling process described above shall be repeated.

h. Upon completion of descaling, the used descaling solution shall be removed by the Contractor, to a designated pit or receptacle on the plant site, and poured into a sufficient amount of sodium carbonate or sodium hydroxide solution to render the mixture neutral.

Lubrication Abstracts

Edited by W. E. Campbell

Selected literature compiled by members of the Technical Committees and Industry Councils of the Society.

W. E. O. Trowbridge (Managing Director, Sharples Centrifuges Ltd.), "Centrifugal Purification of Oils for Marine Service," Institute of Marine Engineers, Jan., 1960, pp. 1-22.

The development of centrifuge types applicable to the purification of marine oils is traced from batch units to continuous self-cleaning machines. A mathematical basis of comparison is suggested for the different types and operating rates of the existing installations. Finally there is a brief discussion for the pretreatment of oils with particular reference to neutralization of acids and a table showing the types of impurities encountered in marine fuels and lubricating oils. *(Abstractor: A. S. Gressel)*

K. Walters, "Motion of Elastico-Viscous Liquid Contained Between Concentric Spheres," *Quarterly J. Mechanics and Applied Mathematics*, v. 13, pt. 3, 1960, pp. 325-33.

The theory is developed of oscillatory flow of an elastico-viscous liquid in a concentric-sphere elastoviscometer, in which the outer of two concentric spheres undergoes forced harmonic angular oscillations about a vertical axis and the inner sphere is constrained by a torsion wire. It is concluded that such an instrument would distinguish different elastico-viscous liquids in much the same way as a coaxial-cylinder elastoviscometer. *(Engineering Index: per W. E. Campbell)*

L. Grunberg and D. Scott, "Effect of Additives on Water-Induced Pitting of Ball Bearings," *Jour. Petroleum Inst.*, v. 46, n. 440, 1960, pp. 259-66.

Using the four ball test, the effect of additives in mineral oils on water-accelerated pitting failure of ball bearings is investigated. The most effective additive found is isoamyl alcohol. An imidazoline derivative is also effective. Oleic acid and triethanolamine are satisfactory under conditions of high supersaturation with water. Water-soluble anti-corrosive and detergent additives have no beneficial action. *(Engineering Index: per W E. Campbell)*

A. E. Brenneman, E. D. Nostrand, "Multigraded Dispersant Detergent Oils—Best Way to Keep Engines Clean," SAE Paper no. 126V, October 1959, 27 pp.

Described is a series of tests conducted with New York City taxicabs for 18,000 to 25,000 miles to study the effect of various dispersants and detergent-inhibitors in preventing sludge, varnish deposits, and engine wear. Both multigrade and straight grade oils were evaluated and the results are presented in charts and graphs. The conclusions drawn from the program are: 1. V.I. improver/dispersants as a class, used in combination with detergent-inhibitors in 10W-30 oil, greatly improve engine cleanliness; 2. A new type V.I. improver/dispersant based on a new polymeric backbone and polar component is very effective in minimizing engine deposits; 3. There is nothing unique about either the polymethacrylate system or the presence of nitrogen in providing dispersance to a polymer system; 4. Multigraded 10W-30 motor oils utilizing V.I. improver/dispersants can give better engine cleanliness than straight-graded oils containing detergent-inhibitors at the Supplement I level of diesel detergency; 5. Certain combinations of V.I. improver/dispersants and detergent-inhibitors may adversely affect other oil properties, so all compounds should be thoroughly tested. It is concluded that a multigraded oil with polymeric V.I. improver/dispersant and a detergent-inhibitor is the best way to keep an engine clean for severe stop and go driving. The Appendix gives details of oils, fuel, and test data. *(Abstractor: George C. Lawrason)*

"Ramrod Lubrication System" Automobile Engineer, v. 50, no. 1, 1960, pp. 24-25.

A new system of bearing lubrication to increase the life of main and big-end bearings of large diesel engines for industrial and marine application is described. The system involves the injection of lubricating oil at high pressure into the bearing shortly before each application of inertia or gas loading. During the development, a test machine for the study of oil films in bearings was devised. With this machine, the oil film is observed in a transparent journal and simulation of bearing loading is effected by two hydraulic cylinders regulated by cam mechanisms to conform with any polar load diagram which can be met in practice. *(Abstractor: George C. Lawrason)*

A. de Faro Barros and A. Dyson, "Piston Ring Friction—Rig Measurements with Low Viscosity Oils", *Institute of Petroleum*, v. 46, no. 433, January 1960, pp. 1-15, Discussion pp. 15-18.

This paper describes a study undertaken to measure the friction forces between a piston and cylinder for the purpose of rating lubricating oils. Using a rig developed by Eilon (Proc. Inst. Mech. Engrs. 1957), in which the air pressure load on the fixed piston and the cylinder speed can be varied, the authors show that the friction forces tend to be greatest at or near the dead center positions and that the coefficient of friction is less at 100 psi gas pressure in the piston than at zero gas pressure. The authors hypothesized that the latter phenomenon is probably due to the rings twisting in the grooves and ring twisting is dealt with in some detail. In all cases the coefficient of friction increases as the severity of operation increases. Values for the friction forces are shown on a scope and representative traces are presented. The relationship between friction forces and percent of stroke and the effects of various ring profiles and oil types are described. Application is made of the data obtained to conditions in an engine for maintaining hydrodynamic lubrication. It is shown that hydrodynamic lubrication would exist for only about the latter 79% of the power stroke. The appendix describes the equipment, oils, and some of the procedures used. A discussion at the close of the article deals mostly with the theory of hydrodynamic lubrication. Thirteen references. *(Abstractor: George C. Lawrason)*

Patent Abstracts

Compiled by Ann Burchick
Aluminum Co. of America

Lubricating Compositions, Patent No. 2,958,663 (L. C. Westcott and H. M. Lurton, assignors to Shell Oil Co.) A lubricating oil composition consisting essentially of a major amount of mineral oil and from about 0.01% to about 5% each of sulfurized oleic acid containing from 10% to 30% sulfur in the molecule, C_{18}-C_{22} alkenyl succinic acid, chlorinated paraffin wax containing from 20% to 60% chlorine in the molecule, diphenylamine and N,N-salicylal-1,2-propylene diamine.

Extreme Pressure Lubricants Containing Molybdic Xanthates, Patent No. 2,951,040 (G. Hugel and R. Courtel, assignors to Institut Francais du Petrole des Carburants et Lubrifiants) A lubricant consisting essentially of a mixture of about 90 to 99.5% by weight of a lubricating mineral oil and about 0.5 to 10% by weight of a molybdic xanthate.

Synthetic Lubricant Composition, Patent No. 2,951,041 (C. E. Saunders, assignor to the United States of America as represented by the Secretary of the Navy) A water soluble lubricating composition consisting essentially of a water soluble alkylene oxide oil of copolymerized ethylene oxide and propylene oxide, and from 0.2 to 5.0 percent by weight of an additive selected from the group consisting of the mono-, di- and tri-ethanolamine salts of an aliphatic acid having from 8 to 20 carbon atoms.

Lubricating Oil Additives, Patent No. 2,951,050 (P. J. V. J. Agius and P. R. Morris, assignors to Esso Research & Engineering Co.) An oil soluble material useful as a detergent additive in lubricating oil which is a partial ester of a polyvinyl alcohol having a molecular weight of 500 to 50,000 and material selected from the group consisting of carboxylic acids and acyl halides having from 8 to 24 carbon atoms per molecule, wherein about 30 to 95% of the hydroxyl groups of said alcohol are esterified and wherein the remaining hydroxyl groups are replaced by cyanoethoxy radicals.

Wax Compositions Comprising 3-15% of a Paraffin Wax with a Melting Point of 125-135 F. and 85-97% of a Paraffin Wax with a Melting Point of 138-147 F., Patent No. 2,951,028 (J. D. Tench, S. W. Ferris and I. N. Duling, assignors to Sun Oil Co.) A wax composition consisting essentially of from 3% to 15% of a paraffin wax having a melting point of from 125 F. to 132 F., a penetration at 100 F. of above about 100, a viscosity at 210 F. of from 36 to 42 SUS and a W.V.P. (water vapor permeability, T.A.P.P.I. Test No. T464 m-45) of from about 3 to 10, and from 97% to 85% of a paraffin wax having a melting point of from 138 F. to 147 F., a penetration at 100 F. of from 19 to 24, a viscosity at 210 F. of from 37 to 43 SUS, and a W.V.P. of from about 1 to 3.

Powdered Soap Lubricant Containing Inorganic Sulfur Salts, Patent No. 2,957,825 (J. A. Henricks, assignor to Devex Corp.) A powdered soap lubricant consisting essentially of about 10 to 75 percent by weight of an inorganic sulfur salt selected from the group consisting of sodium sulfide, sodium sulfite, sodium hydrosulfite and sodium thiosulfate, about 0.2 to 5 percent by weight of a coupling agent consisting essentially of a water soluble soap solvent having a boiling point above water and a hydrophobic group and a hydrophilic group, and the balance by weight of a high titre soap.

Detergent and Wear Inhibiting Mineral Oil Compositions, Patent No. 2,958,662 (J. A. Edgar, R. F. Bergstrom and R. C. Jones, assignors to Shell Oil Co.) A lubricating oil composition comprising a major amount of mineral lubricating oil and a minor but detergent amount of an oil soluble copolymer of an acrylate ester of an acrylic acid and a long chain aliphatic alcohol and a vinyl pyridine in the mol ratio of 1:1 to 10:1 respectively and having a molecular weight of from 50,000 to 700,000 and a minor, but wear inhibiting amount of an oil soluble basic alkaline earth metal aromatic carboxylate.

Silicone Lubricating Oil Composition Containing Ester for Improved Lubricity and Thermal Stability, Patent No. 2,950,250 (M. Z. Fainman, assignor to Standard Oil Co.) A lubricant composition comprising at least 25% of a silicone oil and from about 5% to about 75% of an ester of a straight chain aliphatic carboxylic acid of from about 4 to about 18 carbon atoms and a polyhydric alcohol free of beta-hydrogen.

Extreme Pressure Lithium Greases, Patent No. 2,959,543 (T. W. Martinek, E. W. Brennan and E. T. Fronczak, assignors to The Pure Oil Co.) An extreme pressure, lithium base grease with increased storage stability of extreme pressure properties which comprises a major amount of a mineral oil boiling in the lubricating oil boiling range, and a minor amount of a gelation agent, consisting essentially of a mixture of lithium soaps of group (1) saturated monohydroxy fatty acids having 12 to 22 carbon atoms in the carbon chain, and group (2) sulfurized-phosphorized non-hydroxy mono-unsaturated fatty acids having 12 to 22 carbon atoms in the carbon chain, the group (1) acids constituting approximately 80-20 percent by weight of the mixed acids and the group (2) acids constituting approximately 20-80 percent by weight of the mixed acids of said lithium soaps, the gelation agent being present in a sufficient amount to produce a gelled grease structure, said grease being formed by the in situ saponification with lithium hydroxide of fatty material of the group consisting of glyceryl esters of group (1) acids and glyceryl esters of group (2) acids which have been sulfurized by reaction with elemental sulfur at about 330-340 F. and then phosphorized by reaction with a phosphorous sulfide at about 220-230 F. until substantially all sulfur and phosphorous were chemically bound, as measured by the copper-strip test, in lubricating oil, and followed by formation of a soap-oil dispersion and dehydration of the dispersion at 320-340 F., and cooling the product to produce a homogenous jelled grease structure.

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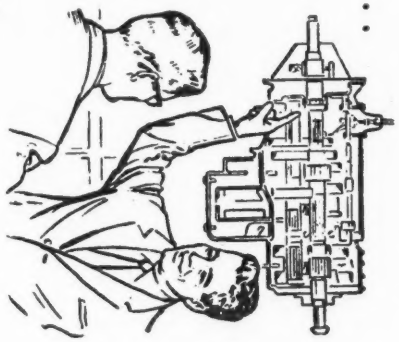

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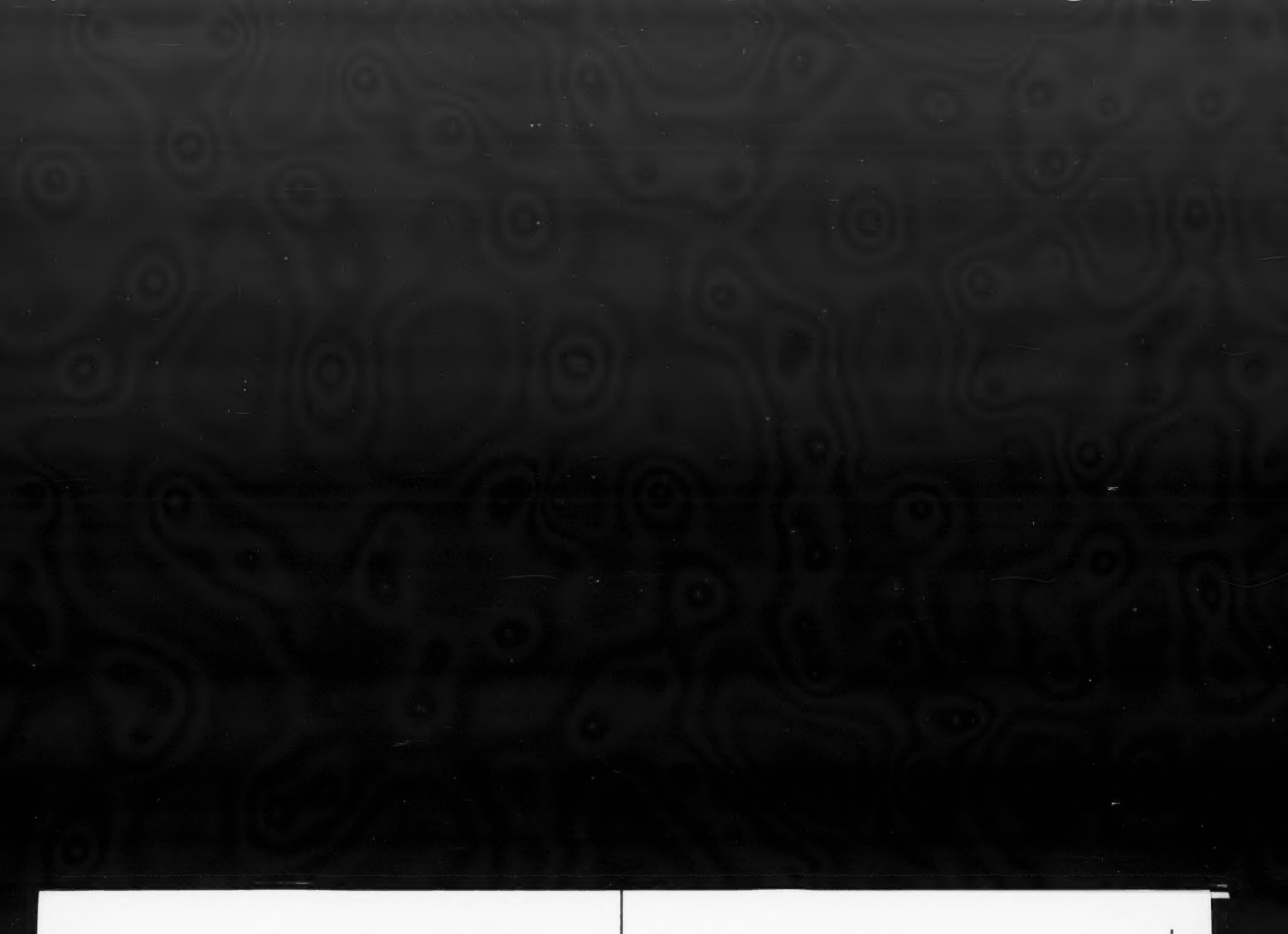

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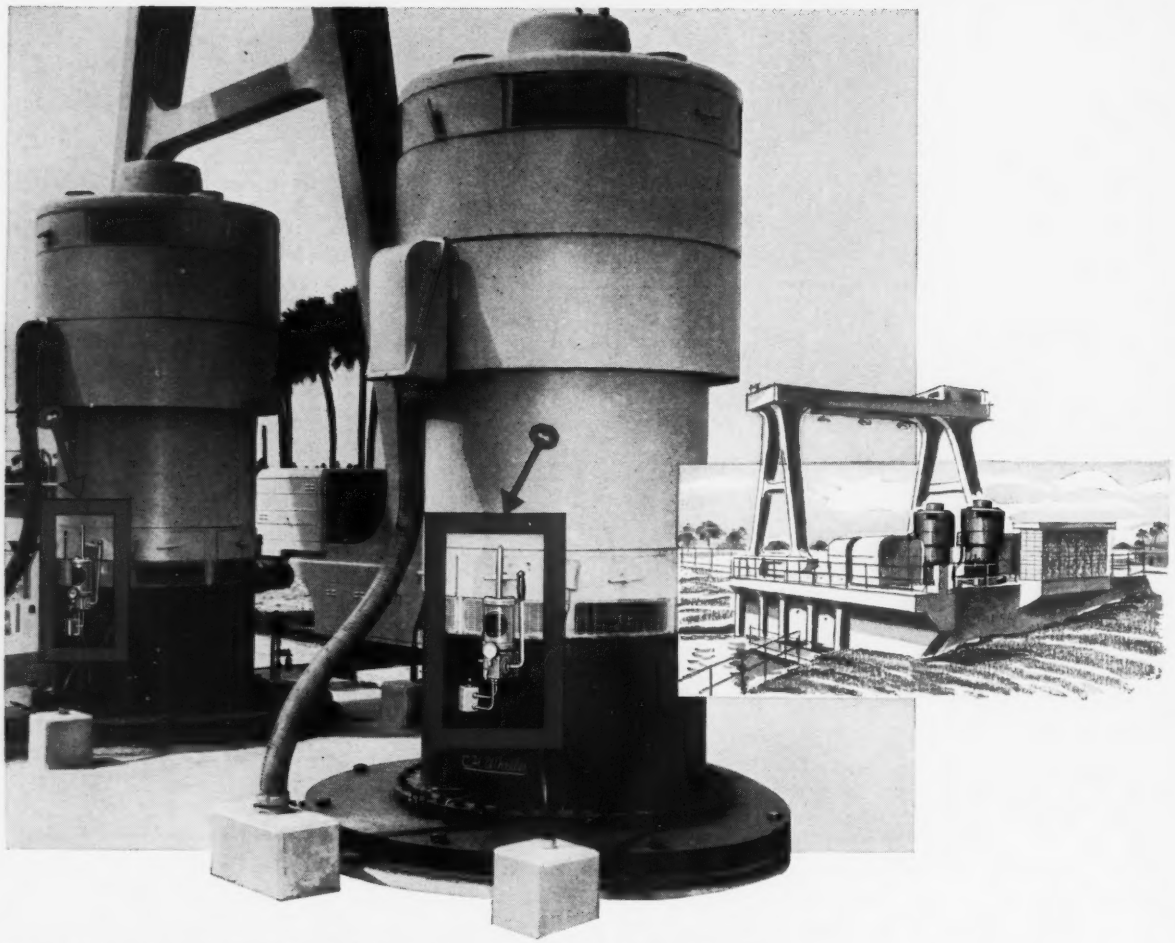

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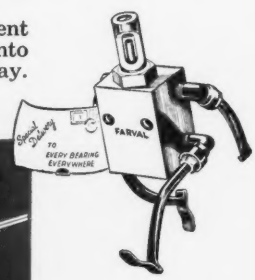

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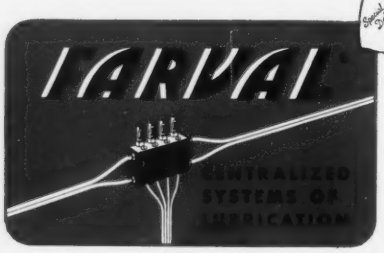

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